International Marine Engineering

JULY, 1908.

NEW JAPANESE TRANSPACIFIC LINERS.

With the sailing from Hong Kong on June 2 of the steamship *Tenyo Maru,* and the approaching completion of her sister ship, *Chiyo Maru,* there has been begun a new service to San Francisco of the Toyo Kisen Kaisha (Oriental Steamship Company). These two ships, and a third now building, are the largest ever built in Japan, and are the first so built to be fitted with steam turbines. They are of about 13,500 tons gross register, and are propelled by triple screws, each actuated by a turbine of the Parsons type. The central screw is operated by a high-pressure turbine with a rotor 76 inches in diameter. The two low-pressure rotors are 106 inches in diameter and operate the side screws. Each shaft has a diameter of 12 1/16 inches. The reversing turbines, which are of ample power to assure efficient maneuvering, are, as usual, incorporated in the low-pressure casings. The turbines are designed for a working boiler pressure of 180 pounds per square inch, and will develop about 17,000 horsepower at a speed of 270 revolutions per minute. This is expected to give the ships a trial speed of 20 knots and a continuous sea speed of 18 knots. The propellers are three-bladed and of small diameter.

THE NEW TURBINE-DRIVEN TRIPLE SCREW STEAMSHIP TENYO MARU.

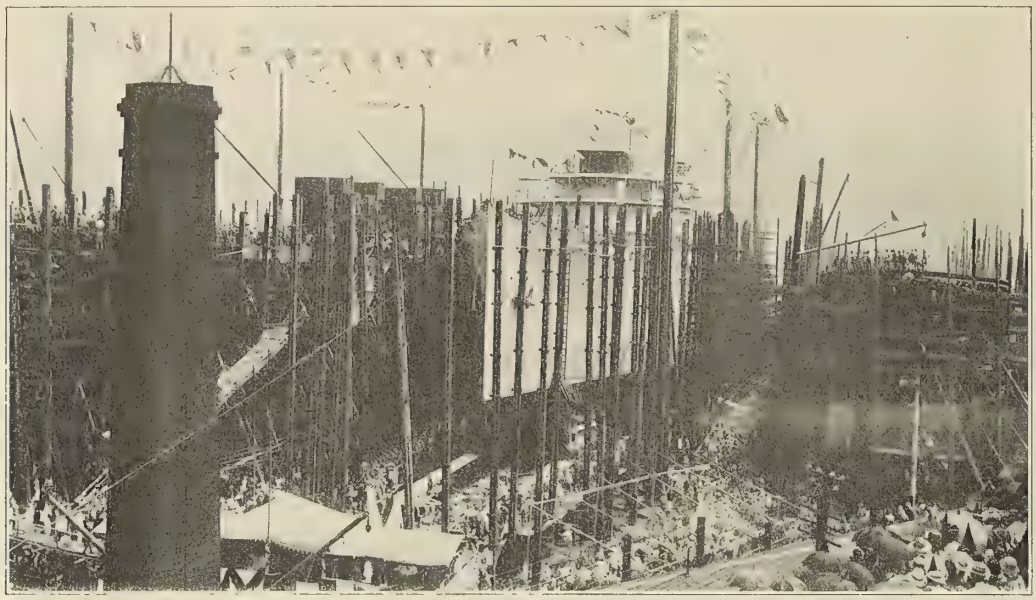
The steam generating plant consists of thirteen single-ended Scotch boilers, with a diameter of 15 feet 9 inches and a length of 11 feet 6 inches. Each boiler has four Morison suspension furnaces, and is operated under Howden's forced draft. The aggregate heating surface is 37,660 square feet, or 2.21 feet per designed indicated horsepower. The boilers are arranged in two compartments, separated by watertight bulkheads, and each set discharges the products of combustion into a funnel of elliptical shape, having major and minor axes 12½ and 9½ feet, respectively. Liquid fuel will be used.

The auxiliary engines in the engine room are in nearly all cases fitted in duplicate as a protection against possible breakdown. There are in all twenty-eight sets of pumps, three evaporators and one duplex distiller. The electric generating plant consists of two 75-kilowatt dynamos driven by compound engines. The wiring is done on the double distribution box system, and arranged so as to be accessible in all parts of the ship.

These vessels have a length over all of 575 feet; a length between perpendiculars of 550 feet; and a molded beam of 63 feet. The depth molded to upper deck is 38 feet 6 inches, and 46 feet 6 inches to shelter deck. The height between these two decks is thus seen to be 8 feet. From the shelter to the promenade deck the height is 9 feet; from the promenade to the boat deck it is also 9 feet. With a maximum draft of 31 feet 8 inches the displacement is 21,650 tons.

Provision is made for 275 first class passengers, 54 intermediate passengers and 800 in the steerage. As these ships are to be operated in large measure in tropical waters and on a long run, the greatest attention has been paid to the arrangement of the quarters for these various grades of passengers. Among other things is the provision of ample promenade space and liberal proportions in the living quarters. The ventilation system is said to be such as to insure fresh air, no matter what the weather. Each room is fitted with an electric fan and electric lights, while the system of heating may be controlled by each passenger in his own state room.

The two upper decks (boat and promenade) are devoted entirely to first class accommodations. The shelter deck is the weather deck, and carries cargo gear at the forward and after ends. Amidships, under a deckhouse 280 feet long, are the



THE LAUNCHING OF THE STEAMSHIP TENYO MARU; THE SHIP LEAVING THE WAYS.

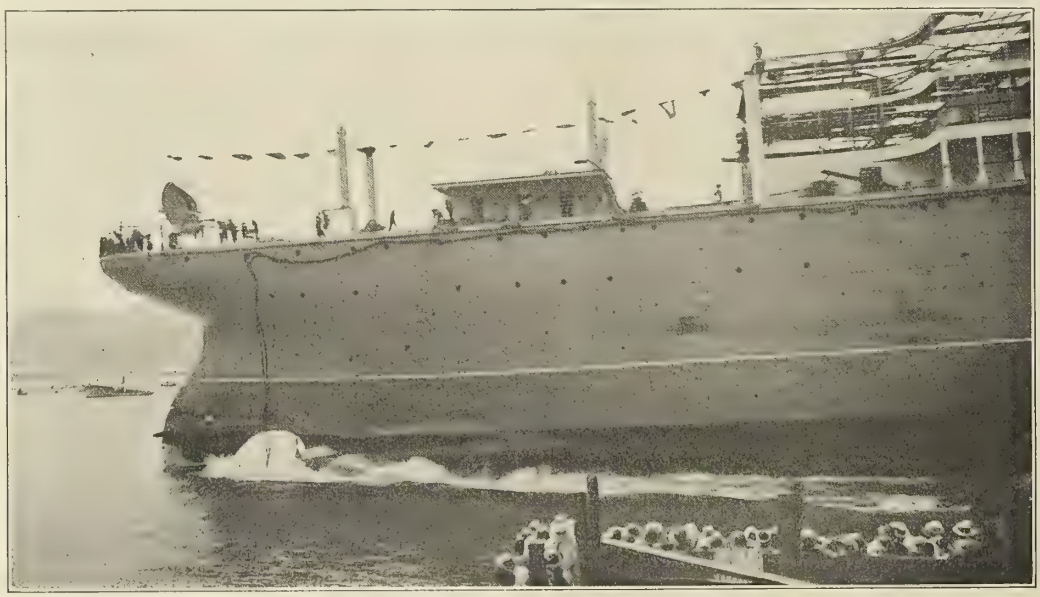
first class cabins and dining saloon. On the upper deck are quarters for the intermediate passengers, with a few first class cabins amidships. The main and lower decks are fitted largely for cargo stowage. The top of the shaft tunnels forms the lower deck and bottom of the after holds.

The first class passengers are accommodated in ninety-six state rooms, in nearly all of which the 9-foot head room rules. On the promenade deck are four suites, containing in each case bed room, parlor, bath room and toilet. In addition to these suites are several so-called family rooms, containing two beds and one sofa, the latter being so arranged that it may be used as another berth. Each of these family rooms is en suite with another room, which can be used as a sitting room, and is provided with ample closet space. Inner rooms have been avoided except on the upper deck, where the state rooms have only one berth. In the furnishing of the rooms, brass and mahogany have been used extensively, and all of the appointments have been designed with an eye to their artistic qualities. All the passageways and alcoves in the first class accommodations are tiled with patent india rubber.

The intermediate passengers are quartered in two-berth rooms, which are simply but comfortably furnished, and are provided with efficient heating and ventilation. These passengers have a large dining saloon, a ladies' room and a smoking room. The Japanese steerage is forward and the Chinese steerage aft on the main deck. The ventilation and sanitation are carefully provided for, and in cold weather the steerages are heated by the thermotank system.

The purser and bureau of inquiry are located on the shelter deck, at the after end of which is a hospital and dispensary. On the upper deck is a printing office, from which will be published a daily paper containing news received by the wireless telegraph equipment. The commissary departments are very elaborate, electricity playing an important part. The refrigerating plant insures a constant supply of fresh food. There are separate galleys for the Chinese and Japanese steerage passengers.

For the carrying of cargo the ship has six holds of nearly equal capacity, reached by eight hatchways. Each hatchway is provided with two winches furnished by Clarke, Chapman &



THE LAUNCHING OF THE STEAMSHIP TENYO MARU; THE SHIP TAKING THE WATER.

THE SHIP AFLOAT.

Company. There are in addition two 25-ton derricks for the handling of heavy weights. Twin capstans are fitted at the forward and after ends of the shelter deck. The anchor cables are 2⅞ inches in diameter, and operate four Hall patent stockless anchors. The anchors and cables weigh 90 tons.

STERN VIEW OF TENYO MARU ON THE WAYS.

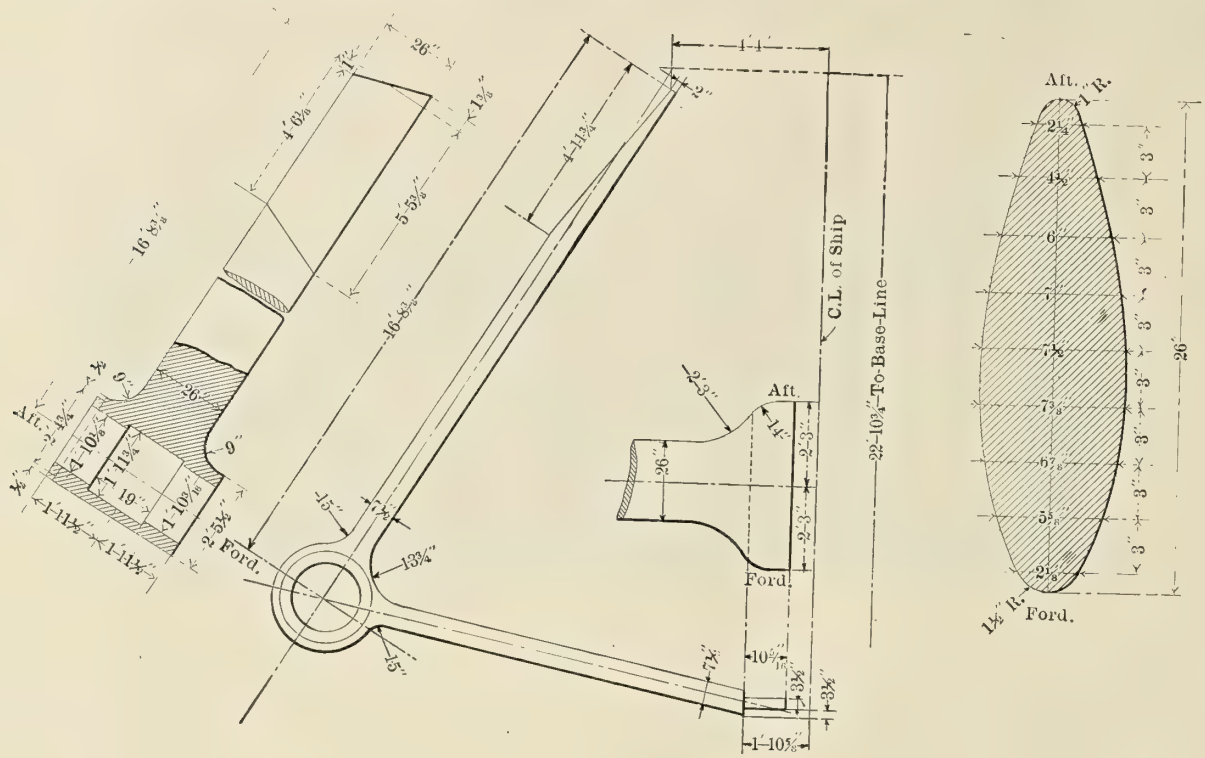
There is telephonic connection between all working parts of the ship. The watertight doors to the numerous bulkheads are installed on the "Long-Arm" system, and in an emergency can all be closed simultaneously from the bridge.

Among the other features might be mentioned a well-equipped gymnasium, a nursery, an auxiliary saloon, where private parties can be given, a dark room, a dancing floor on the after-part of one of the decks, with the piano at convenient command, a lounging room where both coffee and cigars may be enjoyed, and the usual public rooms to be found on all vessels of the present day.

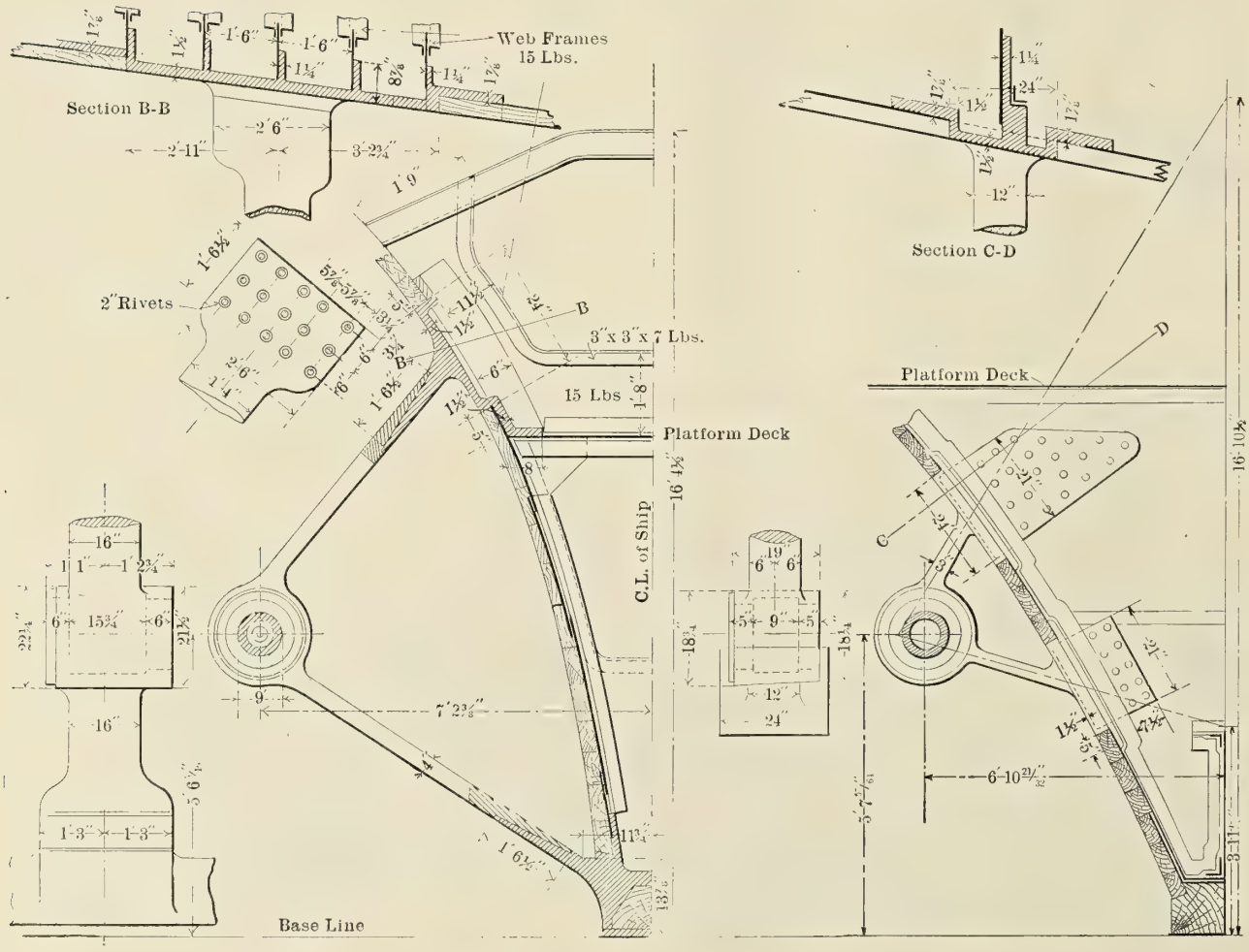
All three ships are products of the Mitsubishi Dockyard & Engine Works, at Nagasaki, and have been built to conform with the requirements of Lloyd's and of the Japanese government (as auxiliary cruisers). The turbines for the first two vessels have been constructed by the Parsons Marine Steam Turbine Company, Wallsend-on-Tyne. The turbines for the third vessel are being constructed by the Mitsubishi Company under license.

The keel of the *Tenyo Maru* was laid in November, 1905, but delays in the transfer of raw materials from England held her back for possibly six months. About 8,000 tons of steel were worked into her before launching, which makes her the heaviest ship ever launched in Japan, or into the Pacific or Indian Ocean. She is the largest merchant steamer launched into these oceans, and also the largest turbine-driven passenger steamer built outside of Great Britain. Each ship has been so designed as to be used as an auxiliary cruiser in case of war, the armament to consist of six 6-inch, ten 3-inch and four machine guns.

The report of the Bureau of Navigation of the Department of Commerce and Labor shows that during the month of April there were constructed in the United States 114 steam and sailing vessels, aggregating 63,176 gross tons. Eight of these vessels, of which five were on the Great Lakes, account for 46,152 tons, or 73 percent of the total. Five of these vessels, including the three largest, were built on the Great Lakes, while two were built on the Delaware River and one at San Francisco. In the total list the steel steamers account for sixteen vessels of 52,217 gross tons, or an average of 3,264. For May the figures were 116 vessels and 51,401 tons, of which ten steel steamers accounted for 43,952 tons.



DETAIL OF THE PROPELLER STRUTS OF THE UNITED STATES BATTLESHIP OHIO.



FORWARD AND AFTER PROPELLER STRUTS OF THE SHEATHED SEMI-PROTECTED CRUISER TACOMA.

A FEW CONSTRUCTIVE DETAILS.

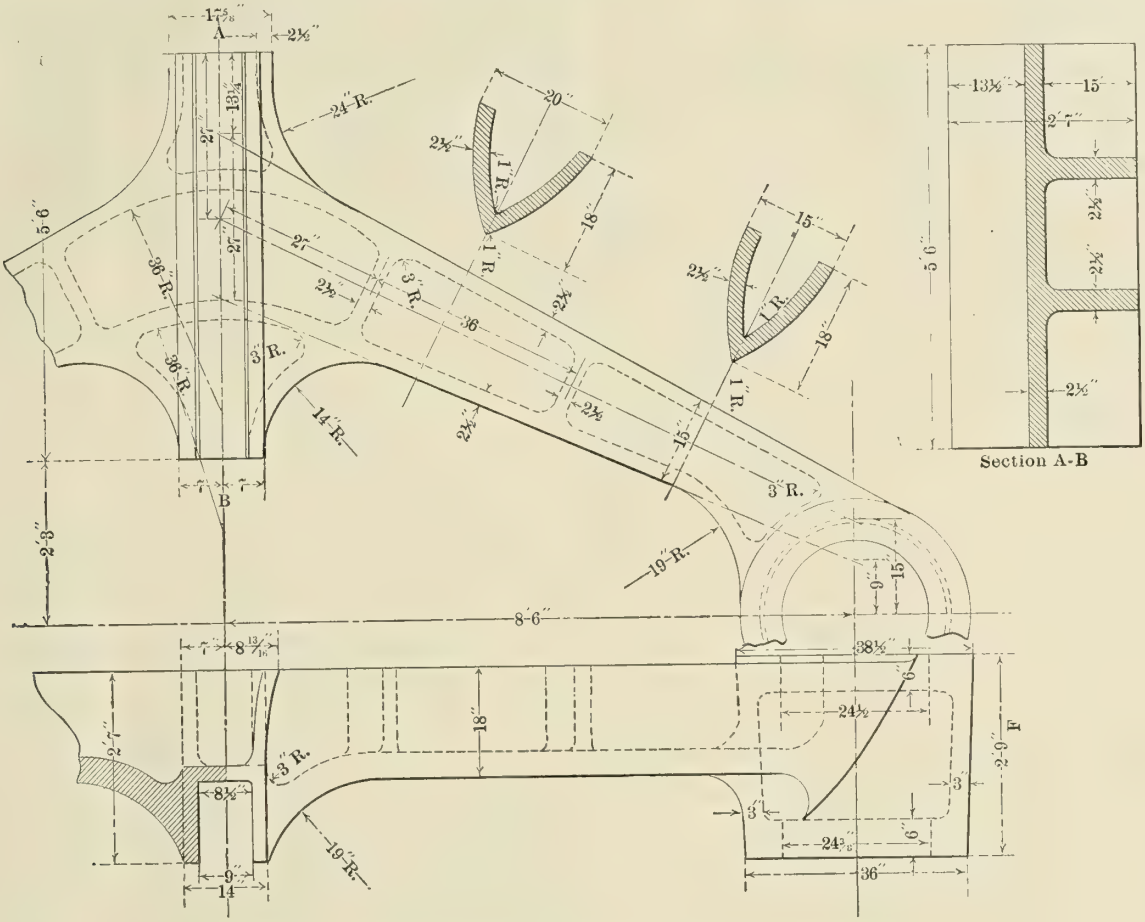
SPECTACLE FRAMES AND PROPELLER STRUTS.

These two items are similar in application, though different in detail. The spectacle frame of the steamships *Mexican* and *Columbian* is made of cast steel, and the rough casting weighed about 21,000 pounds. The center line of the boss for the propeller shaft is 8 feet 6 inches from the center line of the ship, and is 2 feet 3 inches below the bottom of the central part of the casting, which is 5 feet 6 inches deep. The fore-and-aft length of the casting proper is 2 feet 7 inches at the center line, and 2 feet 9 inches at the boss; the intermediate section has a fore-and-aft dimension of 18 inches, the boss has a diameter aft of 36 inches, and forward of 38½ inches.

The after struts on the battleship *Ohio* are of cast steel, and weigh each 18,830 pounds. The center of the boss is 8 feet 3 inches above the base line of the ship, and 13 feet out from the center line. The boss has a length of 3 feet 11 inches. The upper and lower arms of the strut are similar in section, measuring 26 by 7½ inches, as shown in the sectional view. The center line of the lower arm meets the center line of the ship 5 feet 7⅝ inches above the base line.

The struts of the sheathed, semi-protected cruiser *Tacoma* are shown in another drawing. They are of manganese bronze, the after struts weighing 7,623 pounds each, and the forward struts 3,927 pounds each. The after struts have a section in each arm measuring 16 by 4 inches, and the forward struts 12 by 3 inches, each being an elongated egg-shaped form. The center line of the boss at the after strut is 7 feet 2⅜ inches out from the center line of the ship, and 5 feet 6 5/16 inches above the base line. This boss is 2 feet 3¾ inches long. The center line of the boss of the forward strut is 6 feet 10 21/32 inches out from the center line of the ship, and 5 feet 7⅞ inches above the base line. This boss is 19 inches long.

(To be Concluded.)



THE SPECTACLE FRAMES FOR PROPELLER SUPPORT, ON THE STEAMSHIP MEXICAN.

SAIL MAKING.

BY ADRIAN WILSON.

The setting of topsails is almost a chapter in itself, so few realize the importance of knowing how to set topsails. After setting the mainsail, many skippers will loose out the topsail and (1) sheet it out to end of gaff; (2) hoist it up as far as possible, and (3) try to tack it down, which after the first two operations is simply an impossibility and cannot be accomplished. Referring to Fig. 16, remember that a jib-headed topsail is a triangular sail, the same as a jib. The topsail should be set as follows: (1) Hoist sail; (2) tack down—and the tack should always be provided with a good, strong purchase; that is, a reef tackle purchase, or tacked down at the winch, if there is one on the mast for such purposes—and last of all, it should be sheeted out. The sail should be so hoisted and tacked down that when sheeted out the clew is exactly in a right angle drawn from the end of the gaff to the topmast, as indicated by line marked *A* (Fig. 16). If the sail is hoisted so high on the topmast that when sheeted out the clew is above the line *A*, the sheet pulls down the leach, and not only makes a tight leach, but the foot is not taut, and will always flow away to leeward. We very often have complaints that the foot is too long and they cannot get it to set. This is the reason:

If the sail is not hoisted high enough the opposite occurs, and the foot is too tight and the leach slack (Figs. 17 and 18).

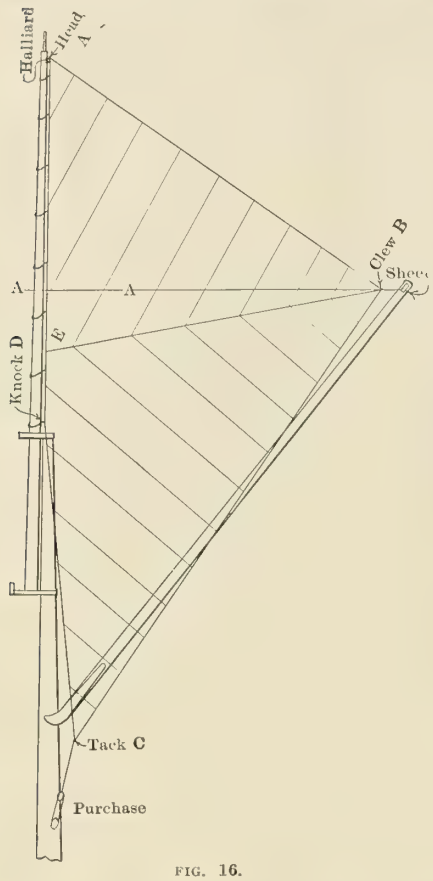


FIG. 16.

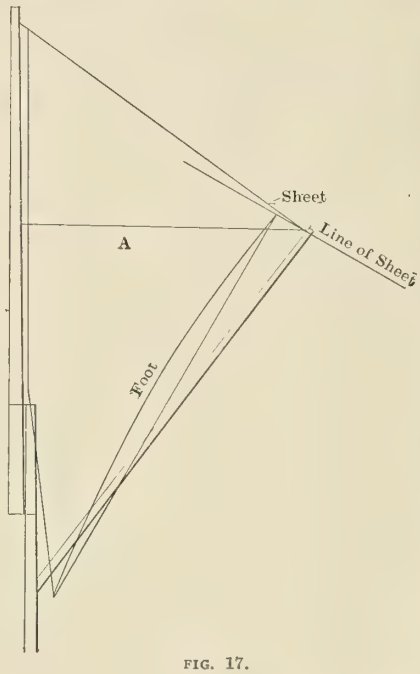


FIG. 17.

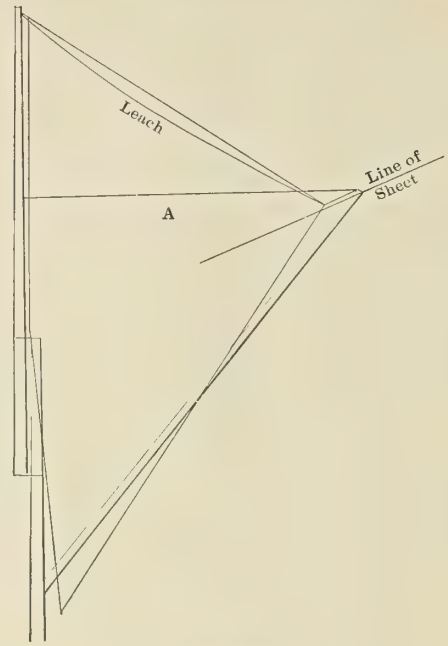


FIG. 18.

If we were to ask a sailing master to haul the jib sheet taut before the jib was hoisted to its place, very probably he would laugh at us and say it was an impossible thing to do. So it is with the topsail. It is a triangular sail, the same as the jib, and should be stretched on its luff before it is pulled out on the sheet. The object to be attained in setting the topsail is to put an equal strain on all three sides of the sail.

It makes no difference if the topsail is cut with cloths parallel to the leach or at right angles to leach and foot, the greatest stretch in a topsail comes on the line *F* to *B*, Fig. 19, as this is directly on a diagonal of the cloth, and gives easily to a strain, so if the sail be sheeted out too far or too hard by the sheet, the clew *B*, instead of being at its designed point, is extended to a point represented by *G*, Fig. 19, and the sail is pulled out of shape. If the sail be hoisted to point *A*, then tacked down to point *C*, when the purchase is put on the sheet, the leach and foot will take an equal strain and the diagonal *F* to *B* will not be over-stretched, and the topsail will fit and set. It is a difficult matter to tell exactly when the sail is hoisted to the proper height on the topmast, and it is better to make several attempts and get it right than go ahead and spoil it and have an unsatisfactory sail or blame the sailmaker. Either go aloft on the mast-head and see that the clew *B* is on a right angle from mast to end of gaff, or, what is better, go off a little distance from the yacht in a small boat, and at a point at a right angle to the yacht, when you can always see if the sail is at its proper height in relation to the right angle. Then you will have a satisfactory sail, and save yourself and the sailmaker a lot of annoyance. The result will more than repay the trouble taken. By doing this you get the position of the distance the tack comes below the jaws of the gaff, after which you can set the topsail without trouble. We have so many complaints that the foot of a topsail will not stand flat, but flow off from the gaff, and it is always caused by setting the topsail too high.

When the sail is new, remember that your mainsail is new. The peak is stretching up and the topsail stretches out. Considerable allowance has to be made in sail for this stretching, and the topsail is considerably smaller than the space it is finally intended to fill, and care must be taken to see that it is set in the proper position at first. The mainsail will stretch up. Consequently, the allowance for stretch on mainsail must be taken into consideration, and whatever the mainsail goes up

THE SCHOONER YACHT TAORMINA IN WINDWARD WORK.
(Photograph by N. L. Stebbins.)

so the whole topsail will go up with it. In addition, some allowance is made for stretch on luff of topsail, so it will not do to hoist the sail two blocks over a new mainsail. (See plan of mainsail and topsail for new sails, with allowance for stretch of both sails, Fig. 20.) Here can be seen the position of these sails as they should be set by the dotted lines, which

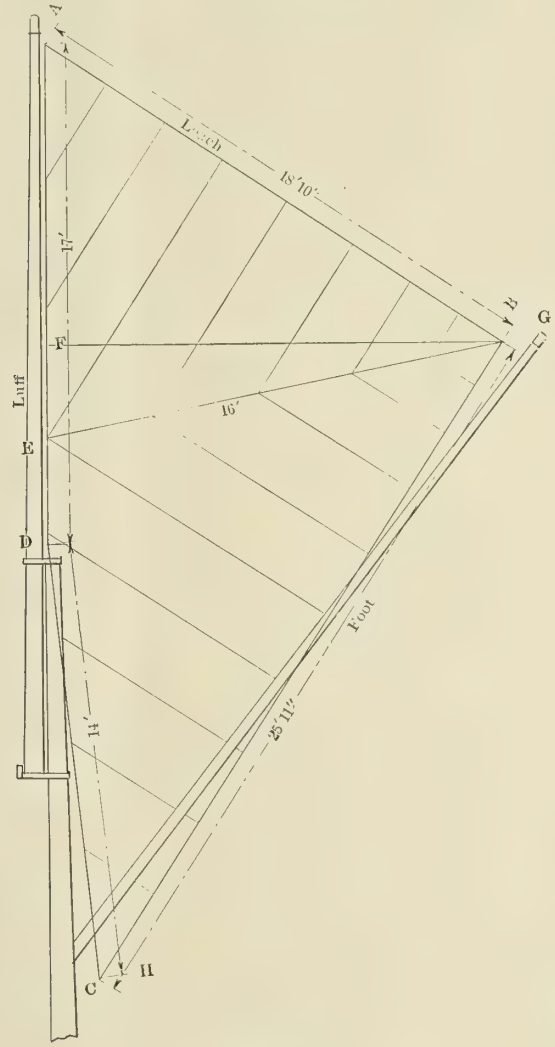


FIG. 19.

represent the new sails, with proper allowance made for stretching.

Now, let us take the foresail. I have taken the rig of a schooner to illustrate the setting of sails. It is a strange fact that, as a rule, the practice is not to set the peak of the mainsail enough, and to invariably overset the peak of the foresail. The foresail is a tall, narrow sail, and the boom and gaff are relatively light spars as compared with the main, and it is a very easy matter to set the peak too high. We want to suggest the following practice in setting the foresail: The luff or hoist should be set to its proper height with gaff hanging at about a right angle to the mast. Refer to Fig. 21, with luff set as suggested above. There should come a fair amount of strain on the diagonal. Suppose that the topping lift has been slacked. Now make the foresheet fast. After the sheet is belayed, peak the sail up until it becomes perfectly smooth and flat in its after part or leach. By setting the sail in this manner you do not overset the peak.

As I have said, the foresail, with its boom and gaff, is light, and it is a very easy matter to overset it. If the sail is set without belaying the sheet, it may be very easily overset, as shown by dotted lines in Fig. 21, and the result will be that the sail, instead of being spread out flat and smooth, will hang from the end of gaff, and the leach will be a perfect bag. Also, the gaff will have a tendency to swing off to leeward, so much so that it will be impossible to make a topsail stand on it without the topsail showing a decided tendency to flap. The gaff, swinging off so far, causes the upper part of topsail to be right in the wind. The over-peaking of foresail also throws the topsail almost completely out, as the distance from mast to end of gaff is thereby shortened (lines *A-B* and *A'-B'*) so much that it is impossible to sheet the topsail. We have often complaints of the foresail stretching up so much that the end of the gaff when tacking fouls the spring stay, when in reality the fact is that it has been simply a matter of setting the peak too high. We have often been called on board a yacht to look at the foresail, and on having the sail properly set we find the end of the gaff will clear the spring stay from 18 inches to 2 feet.

We have, to a very great extent, overcome the difficulty in setting headsails by fitting the luff with steel wire rope in place of the hemp bolt rope. Often the case will be that the leach of a jib is too free and is inclined to whip. If fitted with a hemp luff rope it is more than likely that the sail is not properly hoisted, and often a pull on the halyard will remedy this fault, but the extra pull on the halyard should not be taken until the sheet has been slacked, as it is impossible to get the luff to its right place under these conditions, for the same reason as I have already explained in regard to the gaff topsail sheets. Also of great importance is the lead of the sheet.

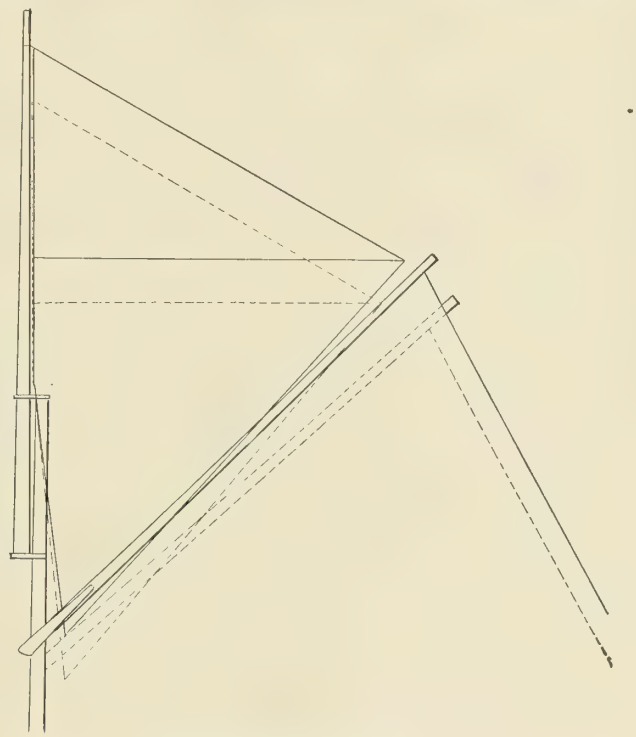


FIG. 20.—FULL LINES SHOW STRETCHED SAIL. DOTTED LINES SHOW SAIL WHEN NEW AND FIRST SET.

Great care should be taken in planning head sails, particularly in regard to location of clew. Head sails have presented the hardest problem that the sailmaker has had to solve. He not only has had to work out the problem of making the sail, but he has also had to work out with the designer the proper proportions that a headsail should be in order to do the best work. My personal experience has taught me that the best headsails are those that are cut with high clews, both in the jib and forestaysail. The position of the clew should be so located that the lead of the sheet will be to a point on deck, or at the rail, that is not only going to be convenient, but also

it must, above all, be so located that the sail will do its best and most efficient work.

On the small-class yachts it is an easy matter to get the proper location of the jib sheet leaders in the following manner: Hoist the jib to its proper height on the luff, and by a temporary sheet fastened in the clew of the sail lead this aft to a point on rail so that you are bringing an exact, even strain on both leach and foot. Now, by going just a trifle further aft with your sheet, you relieve the strain on leach of sail, insuring a good, free leach. The sail when filled with wind will rise a little above the line of the point first found. This plan works in all headsails, but the same idea of the lead of the sheet in setting topsails will not work out in headsails, as the point of contact of the sheet to deck or rail is not on a right angle to the stay.

On larger craft, when it is not practical to find the true lead as above, it will be found necessary to locate the leads from the sail plan. As can be seen by referring to Fig. 22, a plan that works out all right for a forestaysail and jib will not apply to a jibtopsail. The sheet for the jibtopsail will lead at nearly a right angle from the stay, intersecting the clew of the sail. As the jibtopsail sheet can be shifted fore or aft on the deck, this can be adjusted to get both leach and foot to

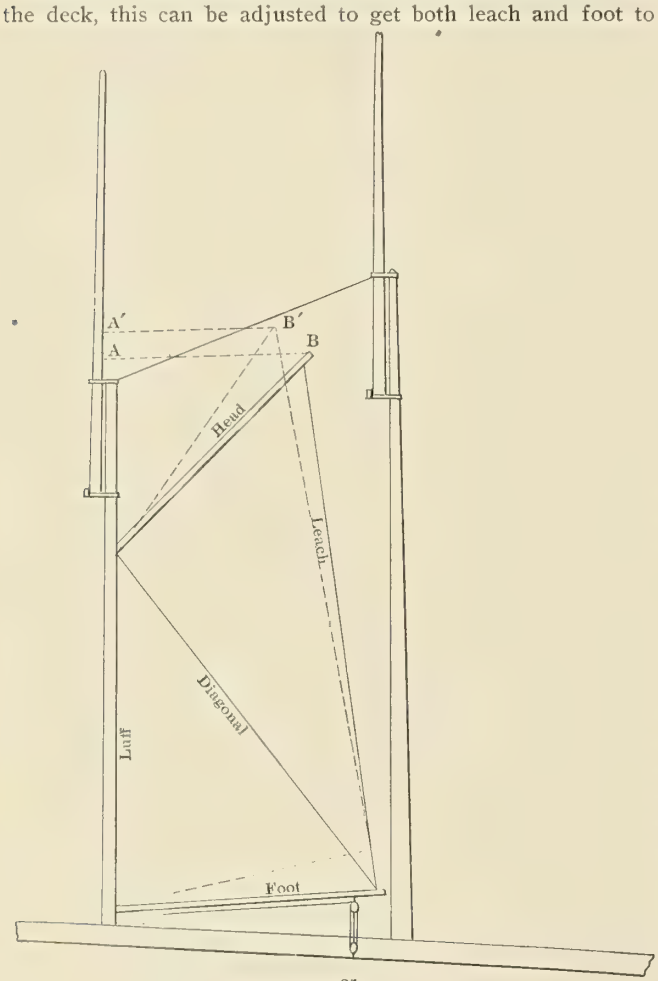


FIG. 21.

stand as desired. The location for forestaysail and jib sheet can also be found by drawing a line on the plan, starting from point *A* (Fig. 22) at the luff, and intersecting the clew of the sail at *C*. This point *A* should be about one-third of the distance of luff from tack. Where this line ends on the deck at *E* will be the correct place for the leads of forestaysail, and where it ends at rail *E* for the jib.

Another important point in headsails is to use great care in designing them, that they will not be too large for the spaces they fill; that is, to avoid having one headsail overlap another. The forestaysail should not be longer on the foot than the distance from the stay to the mast. The clew should not come aft of the mast. In fact, if the clew in tacking swings entirely clear of the mast, the sail will do better work. The jib should overlap the forestaysail only a very little, as the luffs and

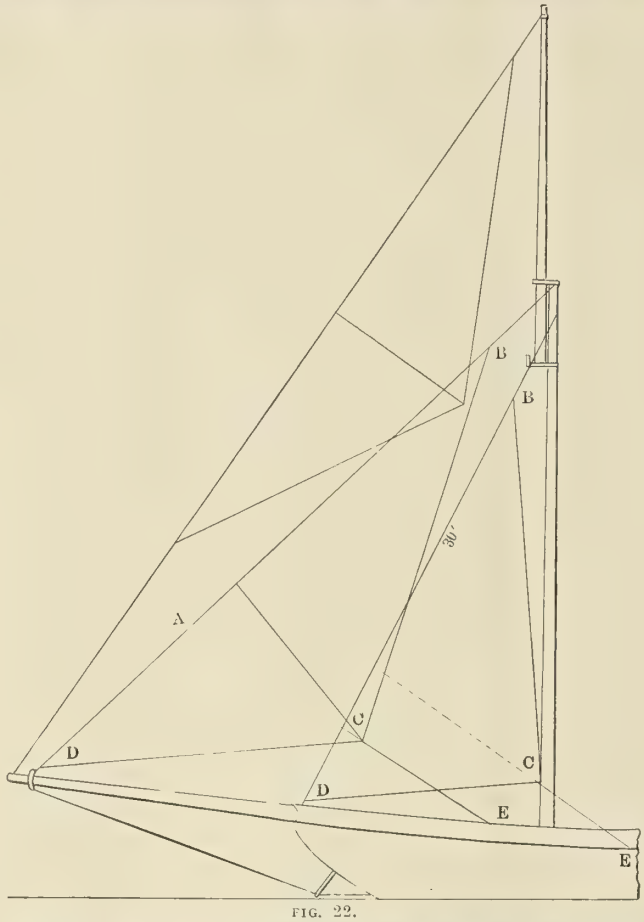


FIG. 22.

leach near the heads or upper parts are coming close to each other, and the wind, passing from the outer sail, will strike the one next aft and cause it to lift from the lee side, which, of course, is to be avoided.

Balloon sails, also, should not be cut too large, especially a balloon jibtopsail. My experience has been that very often the best work of a balloon jibtopsail can be had by not giving it too much lap by the mast, or having its foot cut too low. If this sail is too long on the foot it is impossible to trim it so as to avoid a considerable curve to the foot, which will have a tendency to cause the current of wind to flow back onto the lee side of the after lower sails, a condition certainly to be avoided. Many racing men differ as to the size of the spinnaker; but the majority do not want this too large, and I agree with them on this point. I believe it has been demonstrated in our restricted classes, where the spinnaker can be of only a certain area, as prescribed by the rule, that by having the boom set from 5 to 6 feet above the deck, and the sail hoisted to the top of the mast, and the foot of sail well above water, better results are attained than by using a shorter boom set near the deck, and the foot of the sail close to the water.

So far, what we have had to say about sails and sailmaking has applied only to sails of the fore-and-aft rig. It is not our purpose in this article to go into a description of other rigs, as it is for the benefit of the yachtsman of to-day that we have written what we have on the subject, and the fore-and-aft rig is the only one that will interest him. The fore-and-aft rig has become universal with yachtsmen, on account of its superior windward work, and it is also much handier and more easily managed than other rigs; but there are other rigs that are also

fast for windward work. It must not be taken for granted that the sloop or schooner rig is away and beyond all question faster in windward work than all others. It may be a surprising statement to some who read these lines to learn that a great many square-rigged vessels are remarkably fast in windward work. While the square-rigger may not point as near the wind as the fore-and-after, still there are many that have made remarkable records in windward work. Of the different square rigs in use, such as the full rigged ship, the bark, the brig and the hermaphrodite brig or half-rigged brig, the latter rig has produced in the past many fast-sailing vessels, particularly in windward work.

I have known several of these brigs that in the open sea, where they had the chance to work in tacking, would beat out a fleet of fore-and-afters in a day's work to windward. The plan here given of the brigantine *John McDermott* (Fig. 23) is an instance. This brig has shown marked windward ability, and possibly the position of her spars had a great deal to do with this. In this relation let me call attention to the fact that a square-rigger, when braced up as sharp as possible on the wind, presents an example of our idea of draft in fore-and-aft sails from the very position of her yards in their relation to the center line of the mast, that carries out in the square sail just the same idea that we have tried to illustrate in the fore-and-aft sail. That the upper yards are not braced to such a sharp angle as the lower yards gives the square sail on each mast the same effect or curve as the mainsail or foresail of a schooner.

To illustrate: Suppose we were on the royal yard and looking down on the lower yards in succession, we would see that they are not all braced alike, but as seen in the illustration of the barkentine *Rachel Emery*. Also it will be seen that these sails present a series of curves; and could they be made with the same care and handled with the same skill as the sails on a yacht, they would prove surprisingly fast in windward work. It is a fact that, while the square-rigger will not point so high or so near the wind as the fore-and-after, the reaching quality of her sails produces a very fast vessel, and she makes up in reaching what she lacks in pointing. Herein lays a point that every amateur should note. It is almost a universal fault that the amateur will invariably try to point his boat too near the wind and starve her, whereas, if he would only keep the sails full of wind the boat will every time reach faster, and make the windward work in less time. If the sails are kept full the boat will fairly work out to windward sideways; but if held too high or pointed too near to the wind will make some considerable leeway. The barkentine is also a good rig for windward work. I do not mean to make the statement that the square rig is as good as or the equal of the fore-and-aft; but that many square-rigged vessels are remarkably fast in windward work is the fact. The fore-and-aft rig presents many advantages, it being simpler and much more easily handled, and needs fewer men than the other rigs. There are also the lateen and lug rigs.

The handling and care of sails is a very important item. It has always been the custom to haul sails out on the boom and gaff and make them fast to the spars permanently. The mainsail and foresail on a yacht should be slacked in on the boom and gaff when not in use. If the sail is at all damp or wet, it puts an undue strain on the head and foot ropes, and if allowed to stand in this condition until it dries, or for some time, it becomes stretched, which should not be allowed to happen; so it is safe to slack the lashings at the end of the gaff, and slack up the outhaul on foot when furling the sail, so that it may dry out without being strained. Also, in furling sails it is a custom to roll the sail up tight and snug. This should be avoided. The sail, as it is lowered down, should be folded back and forth across the top of the boom, and secured with the sail stops. The stops should not be put between the foot rope and the boom, but should pass over the gaff and under the boom. If placed between the foot rope and the boom, the stops, when pulled taut, stretch the foot rope, which should be avoided.

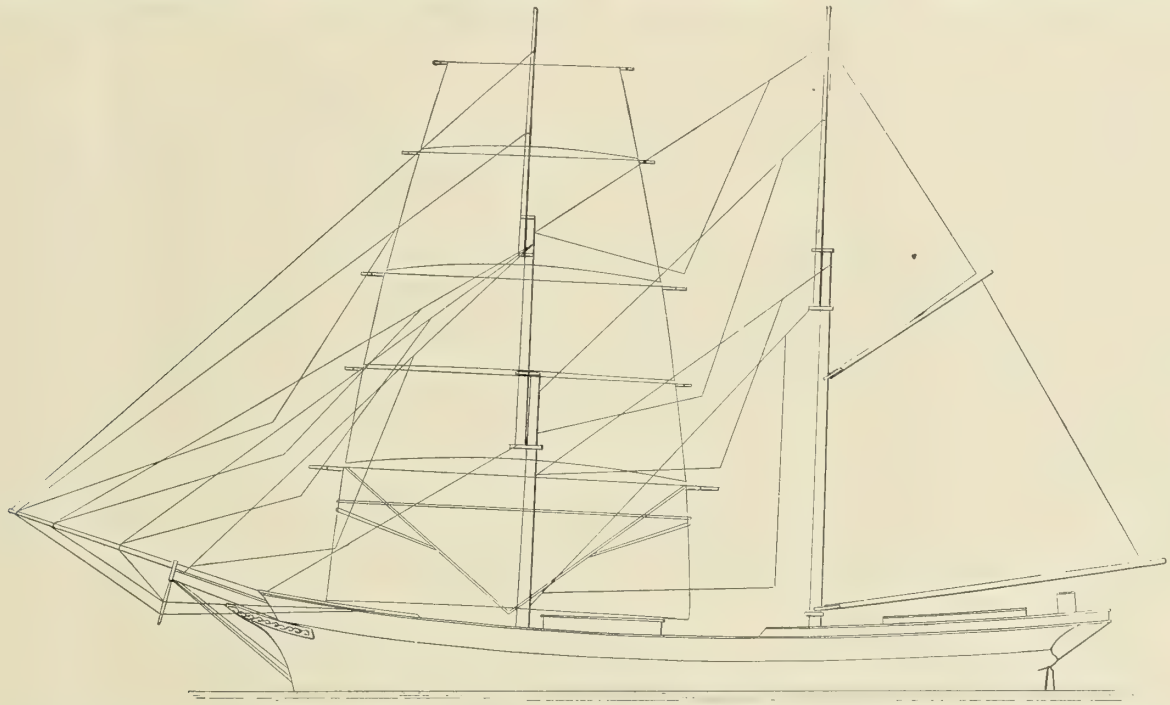


FIG. 23.—SAIL PLAN OF THE HERMAPHRODITE OR HALF-BRIG JOHN MCDERMOTT.

Sails when dried after being wet should be hoisted, if possible, to their exact shape, so that all parts of the canvas will be subject to the same strain. Then all parts will dry with an even strain, and should dry out alike. A great many different processes have been tried to keep sails from mildewing, some of which are very detrimental to the canvas. The best thing for new sails is to soak them in clean sea water; but the sails should be used for several days before doing anything of this kind to them, in order that all parts of the sails may be able to adjust themselves to the strains caused by the actual winds. Great care should also be used as to what conditions a sail is subject to when first used. It is better to exercise great

THE BARKENTINE RACHEL EMERY.

A GOOD SAIL, SHOWING DRAFT IN LUFF. ROSE DOROTHEA.

(Photographs by N. L. Stebbins.)

patience in this and give the sail the benefit of the greatest care, as sail should be set the first time only under the most favorable conditions, and these are a moderately warm day with light winds. Under no condition should a sail be reefed when first set, if we expect to get the best out of it.

For mildew-proofing the following formula will be found to be a very good one and will not injure the canvas: Dissolve 1 pound sulphate of zinc in 40 gallons of water; add 1 pound sodium carbonate. When dissolved, add 2 ounces tartaric acid. This holds the partially separate zinc carbonate without neutralizing the excess of the alkali. The canvas should be soaked in this solution for twenty-four hours. Then dry without wringing.

As I have already said, more poor sails are made on yachts, by improper or careless handling, than are made in sail lofts. So I trust what has been written here may be of benefit to our friends and customers, and, if so, we shall feel that a duty has been accomplished which, while costing us no great effort, may be of value to all who are interested in the sailing yacht—large or small.

Totally Dismasted.

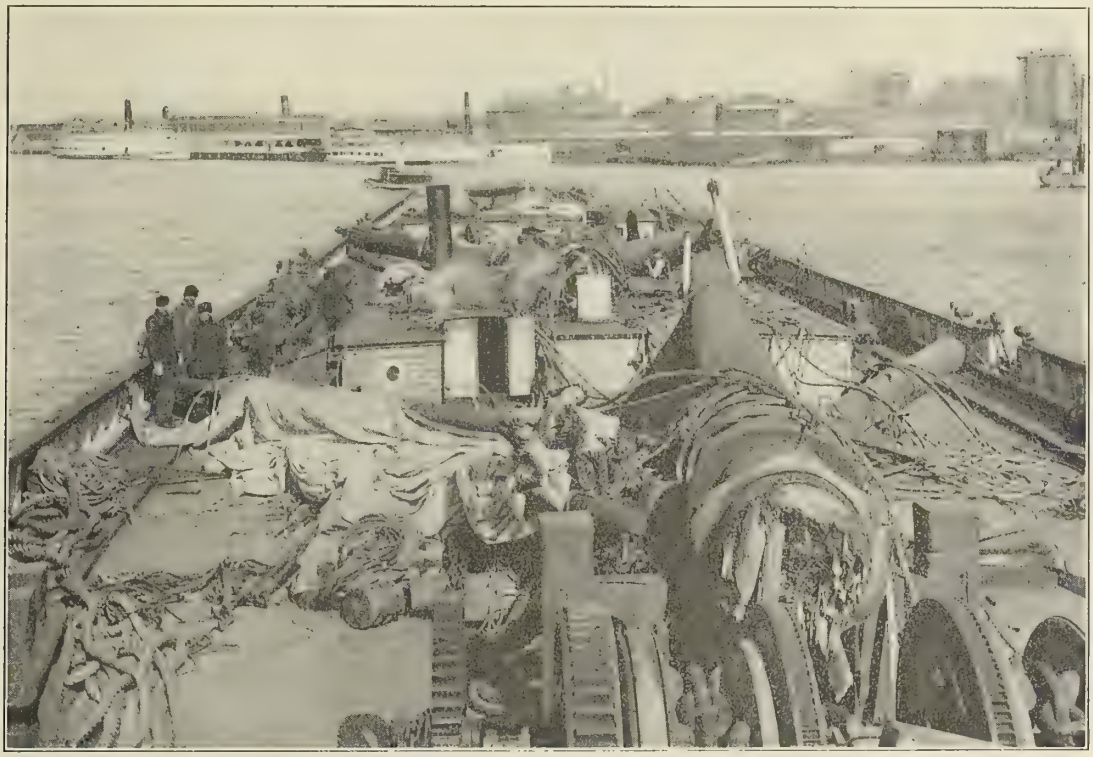
The illustrations depict one of the most unique marine disasters of recent years. They are taken from photographs of the dismasted schooner *Sarah W. Lawrence,* of Manasquan, N. J., just after that vessel was towed into Norfolk, Va., by the United States revenue cutter *Onondaga,* in the latter part of January. The *Lawrence* is a four-masted vessel of 1,301 tons register, 217 feet long, 45 feet wide and 19 feet deep, and carries a crew of ten men, including the captain; she is one of the pioneers of the large present-day fleet of monster coasting schooners, having been built in 1886 at Bath, Maine, and engaged in the coal-carrying trade ever since.

While on her way from an eastern port to Newport News she was struck by the blizzard of Jan. 23, and leveled to the deck, not even the stump of one of her lofty masts projecting above the bulwarks. When the storm struck, her captain, one of the ablest and most experienced of coastwise navigators, shortened sail until only the fore staysail was left, and finally, when it became unsafe to run longer, anchored in twenty-five fathoms of water some 26 miles to the eastward of

THE SARAH W. LAWRENCE WAS TOWED INTO NORFOLK HARBOR TOTALLY DISMASTED.

Winter Quarter Shoal lightship, off the coast of Virginia, and veered all the chain. The vessel was equipped with unusually good ground tackle, which held her even in that exposed position, where the wind had a clear sweep of more than a hundred and fifty miles, and the seas were breaking over her bows, though she was without cargo and high out of the water.

The sails were all furled, the booms lowered to the deck to relieve the masts as much as possible, and every precaution taken to insure the safety of the vessel. The wind and sea were terrific in their violence, and the spray which was flying froze wherever it struck, adding a weight of ice to the already overburdened spars and rigging, until, early on Friday night (the 24th) the vessel pitched so heavily that the rigging could stand the strain no longer, and the bobstays carried away; the fore and main masts fell together, followed the next minute by the two after masts in succession. As the huge sticks fell, they crushed everything in their path until they reached the huge black hull looming above the horizon. Coming up alongside the schooner, a boat was lowered, though a heavy sea was still running, and a hawser taken on board and made fast. Upon going ahead the hawser, an 11-inch manila line, was chafed in two by the jagged pieces of bowsprit projecting from the *Lawrence's* bow. The sea became rougher, and nothing more could be done that night, so the schooner was anchored again, and the *Onondaga* stood by until daylight the next morning, when the line was run again, and again chafed in two. The schooner's starboard anchor chain was then laboriously unshackled, and the hawser made fast to its end. After a few minutes of pulling, the hawser parted and the tiresome work had to be done anew. This time the line held and the *Onondaga* and her tow proceeded toward the Chesapeake. All that day (Sunday, 26th) the wind and sea increased, until by 8 o'clock in the evening it was blowing a strong gale, and the sea was very heavy. Added to this, some of the wreckage hanging over the schooner's stern fouled her rudder,



A SCENE OF DEVASTATION—THE AFTER DECK OF THE SCHOONER SARAH W. LAWRENCE.

solid timbers of the vessel's hull. A great crescent-shaped hole was smashed in the stern, extending down to the transom beam; deck houses were splintered, bitts broken, rails demolished, and the decks covered with an almost inextricable mass of spars, sails, running and standing rigging and wreckage. The two boats which the schooner carried were also destroyed.

Miraculously no one was injured, the crew being all below decks except one man, who contrived to reach the after cabin companionway in time to avoid being struck by the masts in their descent. The masts were securely imbedded in the bulwarks and hull where they struck, and the frightful consequences of having them adrift and rolling about on the decks were thus averted, but, shocked by the terrifying catastrophe and with their lives now dependent only upon the anchor and chain, out of sight of land, at the very height of the stormiest season of winter, theirs was an unenviable lot.

On Saturday, the *Onondaga,* having herself ridden out the gale off the Delaware capes, was proceeding southward, when she sighted the *Lawrence,* a strange-looking craft, with her and it was only by heroic effort that she could be steered at all. As it was, she yawed about in the sea until about 10 o'clock the hawser parted, and the vessel was compelled to anchor again, while the cutter stood by that night, all the next day and the next night, waiting for the sea to moderate sufficiently to permit a boat to be lowered to take the hawser to the schooner.

At daylight on Tuesday morning the surf boat from the *Onondaga* succeeded in running a line to the *Lawrence,* and she was taken in tow once more. From this on, the weather conditions continued to improve, and shortly after dark the vessels passed in the Chesapeake Capes, and the *Lawrence* was anchored in Hampton Roads.

Shipping men in Norfolk, where the schooner was towed the next day, say that she was the most completely disabled craft ever seen in that vicinity, and throngs of curious people visited her while she was at anchor in the harbor.

A certain number of public vessels (revenue cutters) are designated to patrol the stormy parts of the American coasts during the winter months, to render assistance, whenever

necessary, to distressed mariners, and the wisdom of this provision has perhaps never been more strikingly shown than in this case, which has attracted widespread attention, both in maritime circles and elsewhere, reflecting great credit upon Lieutenant At Lee, U. S. R. C. S., who commanded the *Onondaga,* and upon the revenue cutter service. Captain Moore and the crew of the *Sarah W. Lawrence* expressed much admiration for what they called the "bull-dog tenacity" with which the *Onondaga* hung on in the face of many difficulties experienced in effecting their rescue from what was indeed a perilous predicament.

THE HEATING AND VENTILATING OF SHIPS.

BY SYDNEY F. WALKER, M. I. E. E.

HEATING BY ELECTRICITY.

All electrical heating apparatus is based upon the fact that, when an electric current passes through a conductor, heat is liberated in direct proportion to the resistance of the conductor and to the square of the strength of the current. The formula is $H = tRC^2$, where H is the quantity of heat liberated in time t, C is the current strength in amperes, and R is the resistance of the conductor in ohms. The formula may also be written,

$$H = ECt, \text{ or } H = \frac{E^2t}{R},$$

where E is the difference of pressure in volts at the terminals of the conductor. Connection is made with the thermal system by the fact that H is expressed in watts, the electrical unit of the rate of expenditure of energy, and that 17.58 watts equals one B. T. U. This matter is referred to again further on.

Heating takes place in all forms of electrical apparatus, in cables, in the conductors forming part of the coils of dynamos, motors, etc., and also in all forms of electric lamps. But, in the case of cables and conductors forming parts of dynamos and motors, the heat is kept as low as possible, and in the case of lamps, the great object striven for is to obtain as large a conversion as possible of the heat waves into light waves. In heating apparatus the great object to be attained is of course heat, and therefore all electrical heating apparatus is designed with a high resistance, so that as large a quantity of heat shall be liberated as possible, within a given space.

Electrical heating apparatus has so far divided itself into two main branches, the luminous and non-luminous. Luminous electrical heating apparatus is merely an extension of the well-known electric incandescent lamp. The current passing through the filament of such a lamp, it will be remembered, first liberates heat; and if the current is not of a certain definite strength, only heat will be liberated, and the lamp filament remains black, but it is still giving out some heat, though the pressure is too low for the lamp in question. When the pressure and the temperature are a little higher, the lamp becomes red, a larger amount of heat is given out, and the small quantity of light possessed by the red rays. As the pressure and the current are increased, the lamp becomes gradually brighter, finally assuming the well-known yellow tinge, or, if allowed, becoming white hot. In all cases, however, whatever the color of the filament, and whatever the temperature to which it may be raised, the whole of the electrical energy delivered to it eventually becomes heat, and is delivered to the air of the room in which the lamp is fixed.

Under ordinary circumstances, the carbon filament incandescent lamp converts from about 3 to 5 percent of its heat into light, but the light rays are, so far as is known at present, reconverted into heat in the room. The action is the same as the action of the sun's rays upon a greenhouse. It is well known that it is the light rays of the sun which cause the heating effect in the greenhouse, very much more largely than the heat rays. Glass is transparent to light rays, and they pass through the glass into the greenhouse, as through the glass of the incandescent electric lamp, and are converted into heat waves on the other side. In the case of the greenhouse, as glass resists the passage of heat rays through it, the converted light rays cannot escape so easily as they passed into the greenhouse, and the temperature is raised. Similarly, the light rays from the incandescent electric lamp become heat rays on passing out into the room, and the remainder of the electrical energy delivered to the filament becomes heat within the filament globe, and heats the globe in the well-known manner, the heat being then transmitted to the air of the room in the usual way.

Consideration of the two types of heaters, luminous and non-luminous, makes it evident that where *continuous service* is desired, a heater which depends on the setting up and circulation of air currents passing through it gives the best results. If *immediate heat* is required the luminous radiator is practically instantaneous. It heats the person rather than the air in the room, the latter being warmed only indirectly from the heated surfaces on which the rays from the radiator may fall. For immediate, localized heat, for warming the person, it has no superior, and this fact often permits the use of electric heat where it would otherwise be far too expensive. This distinction should be clearly emphasized if an intelligent application of the two forms is to be made.

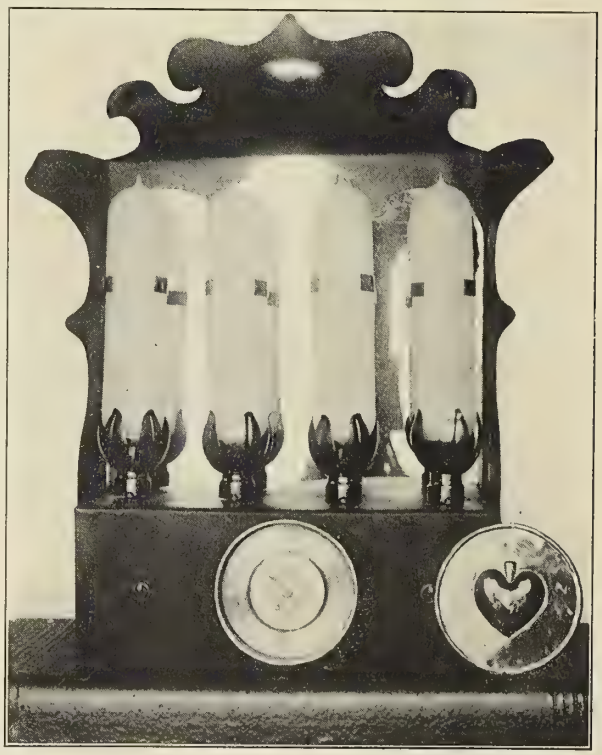


FIG. 19.—GLOW-LAMP RADIATOR AND METER. SAPPLIES, LTD.

ELECTRIC HEATING APPLIANCES ON SHIPBOARD.

Recognizing the vast possibilities in the application of electricity to heating, many manufacturing electric companies have developed a variety of special devices which have already won such favor that it seems certain they will be as commonly used as the incandescent lamp. A ship's lighting plant, usually of more than ample capacity for its intermittent load, offers at once an available source of supply, which, utilized for cooking in the galley or heating in the staterooms, would provide numerous real and profitable conveniences with small increase in cost.

The electric heater is ideal for stateroom use. It is compact and neat in appearance, and easily turned on and off, thus admitting of regulation of temperature for each individual room. It is connected by simple wiring, which is more flexible than steam piping. The wires take little space and can be run anywhere, while steam pipes are bulky and apt to leak, and necessarily heat the spaces through which they pass. It is safe to say that the electric radiator, although deriving its heat indirectly from steam, is no less efficient, when the losses due to leakage and radiation are taken into consideration.

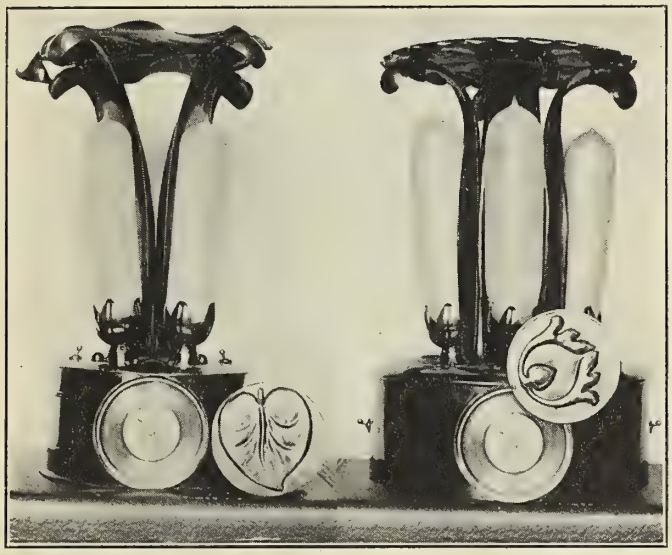


FIG. 20.—GLOW-LAMP RADIATORS WITH METERS ATTACHED.

GLOW-LAMP RADIATORS.

Forms of what are termed glow-lamp radiators are shown in Figs. 19 and 20. They are now well known, and consist of from two to four specially made carbon filament incandescent lamps, usually 9 inches long, with a single horseshoe filament, approximately double the length of the lamp; the two, three

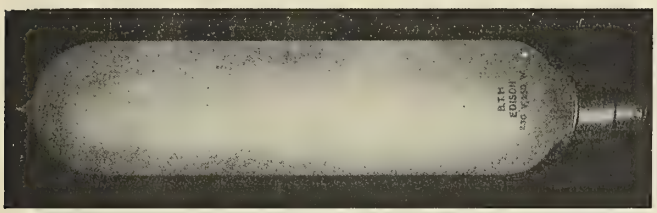


FIG. 21.—BRITISH THOMSON-HOUSTON EDISON RADIATOR LAMP.

or four lamps being held in some ornamental fitting, usually with a reflector behind them, and arranged to throw the whole of the rays from the lamp out into the room. The apparatus is fitted with switches, arranged to connect one, two, three or four lamps, as required, so that the heat delivered to the room may be regulated within these limits. Fig. 21 shows one of the lamps, by the British Thomson-Houston Company, and Fig. 22 shows King Edward's cabin on board the royal yacht, heated by one of Dowsing's luminous radiators.

Some makers are also providing electric radiators with glow lamps, of the pattern described, inside of various inclosures, the appearance being very much the same as that of the non-luminous radiators or convectors described further on. In one form, two or four lamps are inclosed inside a cylindrical copper or brass case, with perforations, the whole apparatus standing a little off the floor. The idea here is that the air passes under the apparatus, up over the lamps, and out through the perforations at the top and the side. This is shown in Fig. 23. Other forms are almost copies of the non-luminous ra-

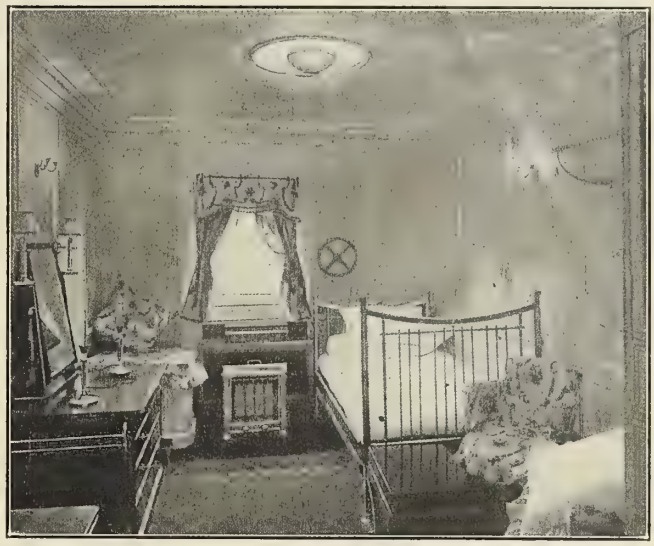


FIG. 22.—KING EDWARD'S QUARTERS ON THE ROYAL YACHT VICTORIA AND ALBERT.

diators. They are rectangular in form and inclose two or four lamps inside a framework, raised slightly from the floor by feet, and the front of the apparatus being closed by slips of ruby glass, the effect is pretty. There are also other forms of this arrangement on something the same lines. The British Prometheus Company has also introduced glow-lamp radiators, in which the lamps are maintained at only red heat.

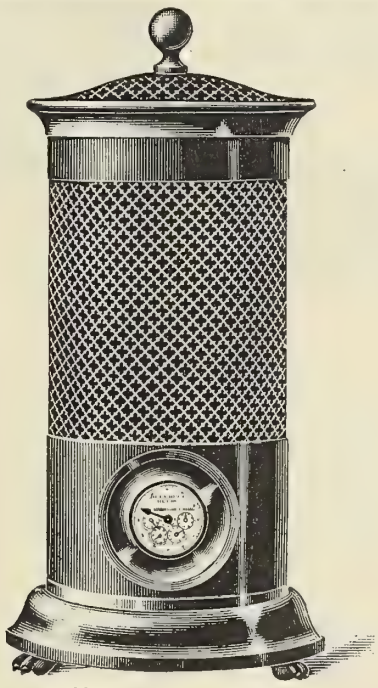


FIG. 23.—GLOW-LAMP RADIATOR.

THE EFFECT OF THE LIGHT RAYS.

It should perhaps be mentioned, *en passant*, that it is claimed by makers of glow lamp radiators that the light rays issuing from the glow lamps have an important office, and, in the writer's view, this is strictly correct. It will be remembered that white light is made up of the different colors forming the solar spectrum, as we see it in the rainbow, and that the rays forming the different colors have different wave lengths, different periods and different properties. Thus, the red rays have comparatively long waves, about double the length of the violet rays, and their property is principally heating. The violet rays at the opposite end of the spectrum have

comparatively short waves, and the principal property is actinic or chemical. It is the violet rays which are most useful in photography.

Between the violet and the red is a long range of rays of different colors, whose properties vary, most of them having been thoroughly worked out. Apparently the yellow rays are those which do most in the direction of furnishing light. Glass and some other substances, the human skin of the white man being one of them, according to some experiments that have been made, are apparently transparent to the yellow and green, and some of the other waves at that end of the spectrum, the waves after passing through the glass or the skin being transformed into heat waves. This has been mentioned as the cause of the heat produced in greenhouses when the sun is bright.

Mr. Dowsing's work also in connection with the use of the electric glow lamp, of the type described for heating, in connection with therapeutics, has shown that the light waves have a very important effect upon the human body. The electric light bath is now well known, and its effects are produced, it is believed, by the light rays issuing from the lamps, and

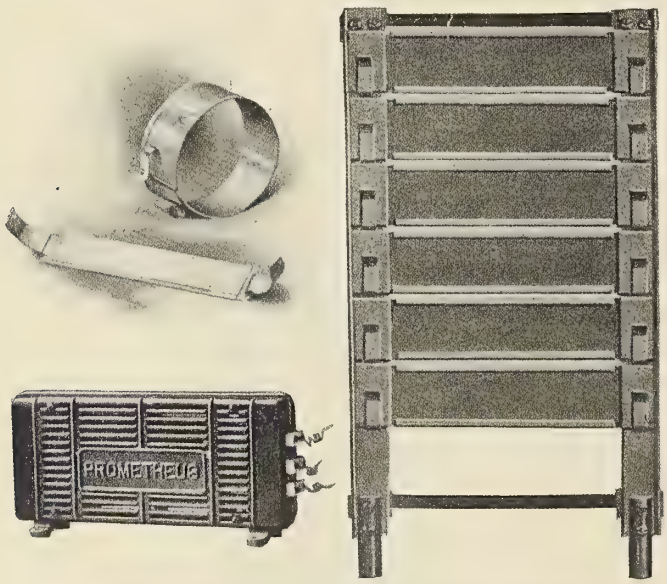


FIG. 24.—PROMETHEUS STATEROOM HEATER, ELEMENTS AND DETAILS.

not by the heat rays. Another peculiar feature about them is, according to Mr. Dowsing, that the pigment under the skin of the black man is not transparent to the light rays. Thus, one cannot give a black man an electric light bath. It does him no good. This would apparently be the reason why the black man can stand the sun's rays. It is not the heat rays which trouble the white man so much as the light rays, which pass through his skin, there becoming heat; while they do not pass through the pigment in the black man's skin.

It will be seen that this has an important bearing upon the question of the warming of living rooms, whether on shore or afloat. Everyone is familiar with the prejudice, as it is thought to be, in favor of a bright, glowing fire; and it is not the fire of red coals that is liked, but one in which white or yellow flames are dancing around the grate bars. If the above reasoning is correct, this tendency, like so many others, is well founded, and the light rays have an important function in the matter of heating. If so, also, the luminous radiator should have an important office to fulfil in the problem of heating saloons, cabins, etc. In the writer's experience, when away at sea, nothing is more pleasant than a bright light in the mess or in one's cabin, and a bright radiator will probably have the same effect.

The lamps in question consume one-quarter of a kilowatt per hour. That is to say, with the usual 100-volt service employed on board ship, each lamp would take 2½ amperes; a radiator of two lamps, suitable for a small cabin, 5 amperes; one of four lamps, suitable for a larger cabin, 10 amperes. The question of the quantity of heat liberated by the radiators will be dealt with further on.

NON-LUMINOUS HEATING APPARATUS.

In the other forms of electric heating apparatus, which are very numerous, conductors, or, as it would probably be more

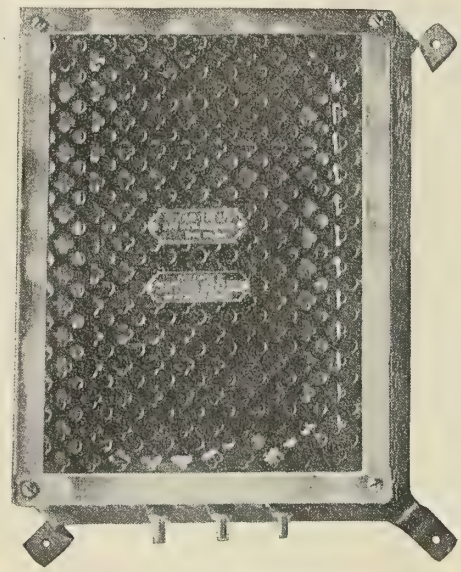


FIG. 25.—PROMETHEUS HEATER, WALL TYPE.

correct to call many of them, semi-conductors, are arranged in various forms, so that electric currents can be delivered to them, and so that the conductors, or semi-conductors, can deliver their heat to the air surrounding the apparatus.

One well-known form of non-luminous radiator, made both in America and the United Kingdom, is known as the Prometheus. It consists of strips of mica, upon which a conductor has been deposited in a layer or film. The strips of mica are provided with clips at the ends, in connection with the powdered conductor, and these clips form the connection to the source of current. Fig. 24 shows the heating elements. The mica strips with their clips, which are called heating elements, are built into various forms of apparatus, known as "convectors," some of which are shown in Figs. 25 and 26.

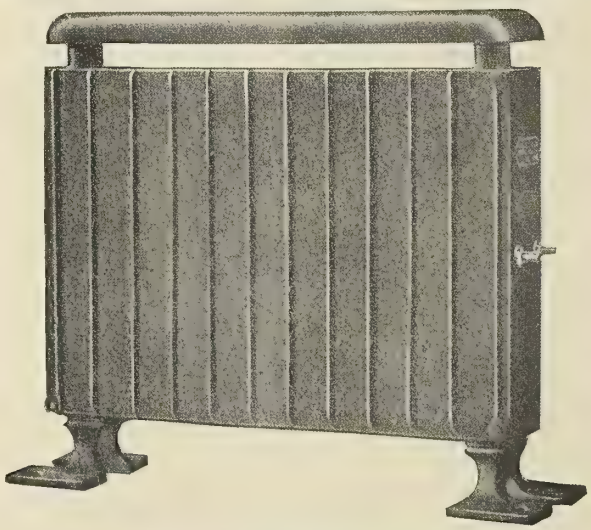


FIG. 26.—PROMETHEUS HEATER, FLOOR TYPE.

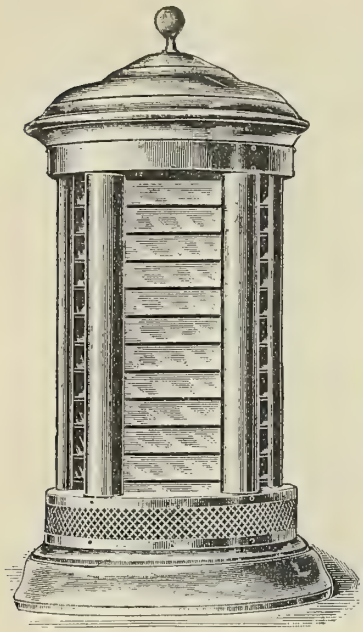
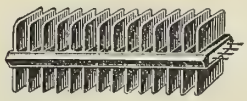
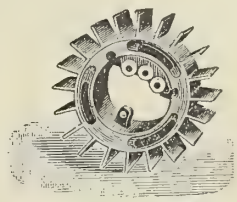


FIG. 27.—PROMETHEUS CONVECTOR.

Unit with Ribs on one side only



Unit with Ribs on each side.



Circular Unit with Ribs.

Unit with Smooth Surface

FIG. 28.—ISENTHAL'S HEATING ELEMENTS.

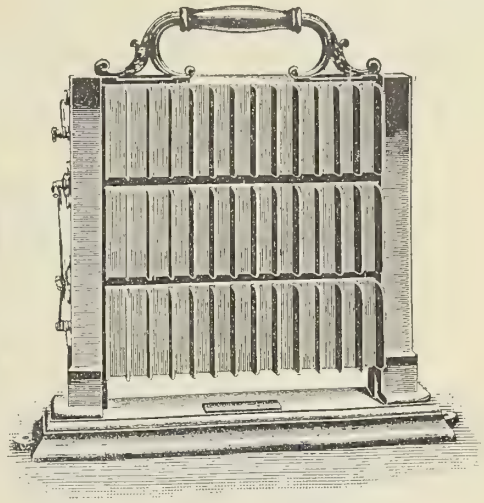


FIG. 32.—ISENTHAL FLOOR-HEATING APPARATUS.

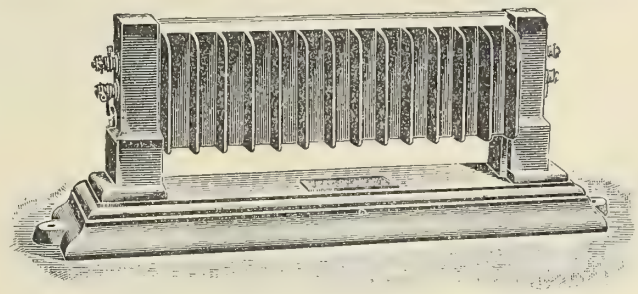


FIG. 31.—ISENTHAL FLOOR-HEATING APPARATUS.

In the usual arrangement there are two metallic uprights, forming the connection to the supply service, and the heating elements are bridged across between the uprights, the whole being inclosed inside of some ornamental arrangement, which may be cylindrical, rectangular or any other convenient form, and which usually has either perforations in the body and at the top, or the equivalent. The whole apparatus stands a little off the floor, the air passing under, up over the heating elements, and out into the surrounding atmosphere.

Another form, made by Messrs. Isenthal, consists of metallic resistances, inclosed within a substance which is an insulator, and which is also highly refractory. The metallic resistance is arranged to have low coefficients, both of expansion and of increase of resistance, so that there may be no change in the form of the heating elements when in use. The heating elements are sometimes made with ribs, as shown in Fig. 28, and there built up into circular or rectangular forms, as shown in Figs. 29 to 32, and inclosed in various ornamental devices, the arrangement being the same as that of the Prometheus. Other firms have other substances. Messrs. O. C. Hawkes, Ltd., London, have a special wire which they claim will stand a temperature of 1,000° F. The Simplex Electric Heating Company, Cambridge, Mass., uses conductor embedded in white enamel, which is fused at high temperature, the enamel providing the insulation and also being very refractory. The

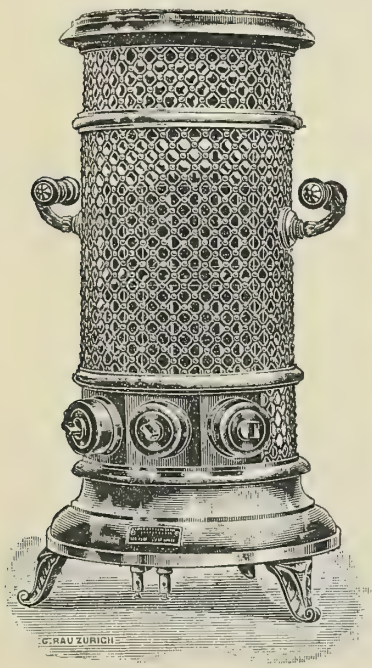


FIG. 29.—CIRCULAR ELECTRIC HEATER.

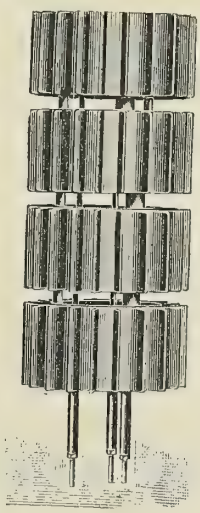


FIG. 30.—BATTERY FOR SAME.

Cartridge Unit.

Quartz Enamel Unit.

FIG. 33.—GENERAL ELECTRIC HEATING ELEMENTS.

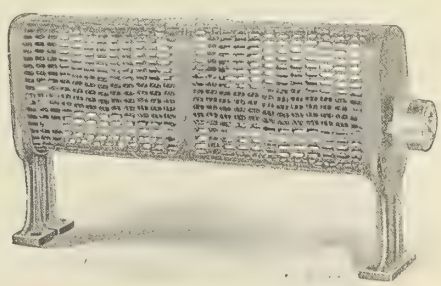


FIG. 34.—GENERAL ELECTRIC STATEROOM HEATER.

General Electric Company, Schenectady, uses a high resistance conductor, coiled into various forms, and covered with a highly-resisting quartz enamel. Forms of these heating elements are shown in Fig. 33, and a stateroom heater in Fig. 34.

(To be Continued.)

THE AMERICO-ITALIAN STEAMSHIP ANCONA ON TRIAL.

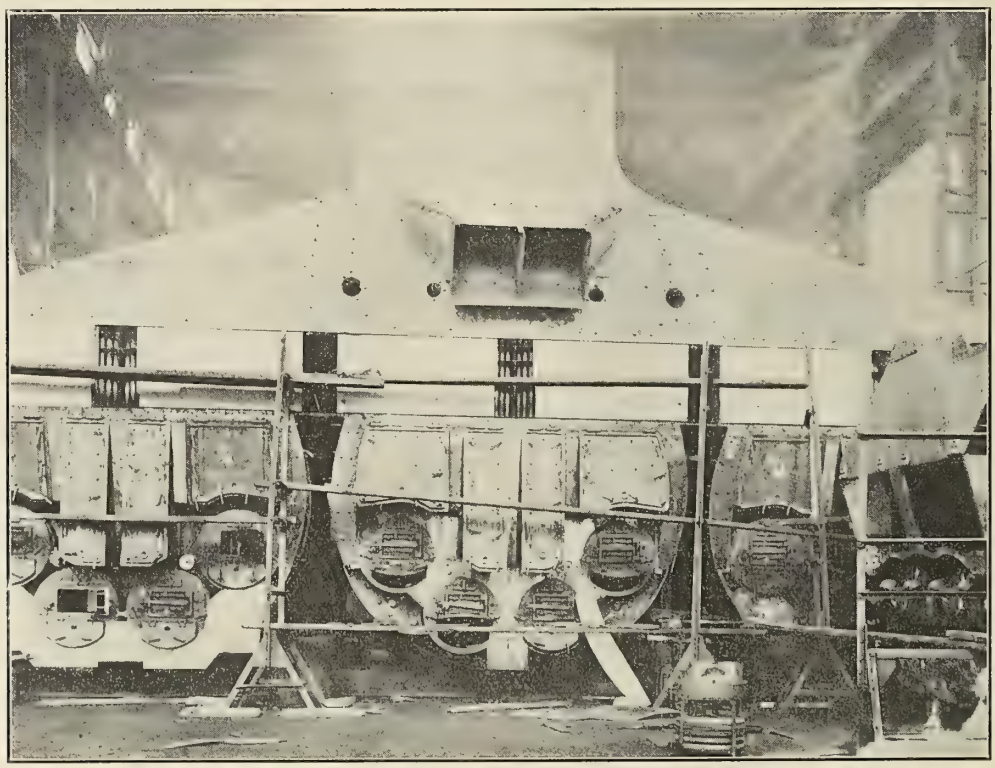
THE AMERICO-ITALIAN EMIGRANT STEAMER ANCONA.

BY BENJAMIN TAYLOR.

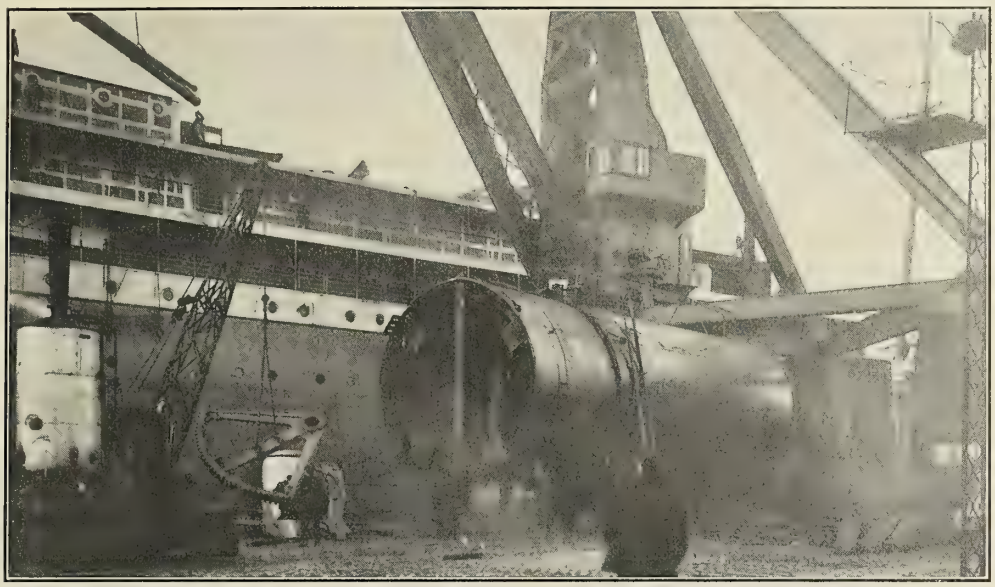
Early in January, Workman, Clark & Company, Ltd., launched from their north shipyard, Belfast, the first of two steamers built by them for the Italia Steam Navigation Company, of Genoa. As the vessel left the ways, she was named *Ancona*. The new ship is 500 feet in length, has a gross tonnage of 8,900, and has been specially designed for passenger service between Genoa, Naples, and New York. She has been built under the special survey of Lloyd's and the Registro Italiano for the highest class in their registers, and also fulfils all requirements of the Italian mercantile marine, and of the United States Department of Commerce. In design and arrangement, special attention has been given to the requirements of the American emigrant trade. The vessel will be one of the most complete in this service.

Accommodation is provided for over sixty first class passengers in staterooms in the promenade deckhouse, with a large dining saloon, with music room and lounge adjoining, placed at the fore end of the deck house. Emigrant accommodation is arranged in the bridge deck house, in the poop, and on the main and lower decks forward and aft. The berths are fitted in blocks, two high, and accommodation is provided for about 2,500 persons. Dining spaces for emigrants, furnished with strong tables and forms, are arranged in the bridge space and on the main deck amidships. The sanitary arrangements throughout the vessel have received special consideration. A special feature is the ventilation of every compartment, both by natural and artificial means.

A complete installation of steam heating is fitted throughout



THREE FOUR-FURNACE BOILERS AND UPTAKES, ITALIAN EMIGRANT STEAMER ANCONA.



HOISTING ABOARD THE RUDDER QUADRANT AND A SECTION OF FUNNEL. STEAMER ANCONA.

the first class accommodation, captain's, officers', and crew's quarters. A large space has been set apart on the lower and orlop decks aft for the storing and preservation of fresh meat, fish, butter, eggs, and milk, and there is an efficient installation of refrigerating machinery.

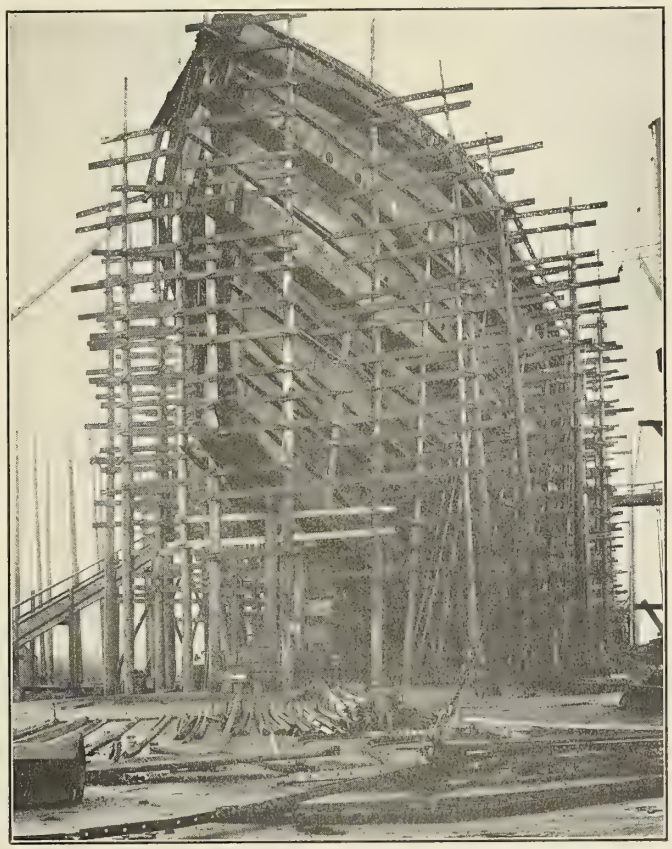
The propelling machinery consists of two three-crank triple-expansion engines, driving twin screws. The cylinders are 26, 43 and 71 inches in diameter, respectively, and the stroke is 4 feet. Internal liners are fitted to both the high-pressure and the intermediate-pressure cylinders; and the valves for controlling the steam and exhaust to and from these cylinders are of the piston type, with adjustable packing rings. The low-pressure cylinders are fitted with double-ported slide valves and improved balance pistons. All the hand gear for controlling and starting the engines is conveniently grouped, and arranged so as to be manipulated from the starting platform. Direct-acting engines of the steam and hydraulic type are provided for reversing the main engines, which develop 7,500 horsepower.

The condensers are built independent of the main engines, and contain an exceptionally large cooling surface, to deal efficiently with the exhaust steam passing from all the machinery when trading in tropical climates. Two large centrifugal pumps, by the builders, are provided for circulating the necessary cooling water through the condensers. Edwards' patent air pumps, with the necessary feed, bilge, and sanitary pumps, are worked by levers and links off the main engines, the two latter pumps being of specially large capacity. The propellers are of the built-up type, and are fitted with blades of manganese bronze.

The feeding of the main and auxiliary boilers is effected by two of Weir's patent automatic feed pumps, working in conjunction with a large feed-water heater of the same make. All the feed water, in passing to the boilers, is properly filtered. Independent pumps of Weir's make are provided for the special duties and requirements of the ship, including a large ballast pump, a fresh water pump, an auxiliary feed pump, besides separate pumps for the sanitary, wash-deck, fire, and other services.

The machinery for supplying electric current for lighting the vessel, and for other motive purposes, consists of three sets of vertical compound engines and dynamos of large capacity, the engines being direct-coupled to the dynamos. In connection with the insulated spaces for provisions, etc., a large refrigerating plant is fitted up in the main engine room, supplied by J. & E. Hall, London, and worked on the CO_2 principle. All the auxiliary plant is arranged in the main engine room, with the exception of the electric installation, which is arranged in a separate compartment on the lower deck, and is easily accessible from the engine room.

There are three large double-ended main boilers, and two auxiliary boilers, all arranged to work at a pressure of 200 pounds per square inch. Each of the main boilers has eight furnaces, provision being made to supply air under pressure on Howden's forced draft system. Two large fans are arranged in a suitable recess in the cross bunker for this purpose. The stokeholds are well ventilated, and are fitted with all the latest appliances for expeditiously discharging the ashes overboard.



BOW VIEW OF THE ITALIAN STEAMER ANCONA ON THE STOCKS.

Special tramway lines and tipping bogies are also provided, for rapidly conveying the coal from the bunkers, for firing purposes.

The five holds, into which the cargo space is divided, are spacious and free from obstruction, the hatches are large and easily accessible, the winches are numerous and powerful, and the derricks, swung from the masts and derrick posts, are plentiful, thus making it possible to handle a full cargo in the shortest possible time, and with the least amount of labor. The life boats are handled by Welin quadrant davits.

THE ANCONA ALONGSIDE CRANES FOR INSTALLATION OF BOILERS AND FUNNEL.

MARINE ENGINE DESIGN.*

BY EDWARD M. BRAGG, S. B.

The object of any heat engine is to reduce the temperature of the working fluid, this reduction of temperature being caused by the conversion of potential energy into work. In some respects the working fluid may be likened to a certain weight of water falling through a certain height. A turbine may be placed at the foot of the fall, and all of the energy of the water converted into work in one stage. A second turbine may be used half way down the fall, and the tail water from this turbine used to run the turbine at the foot of the fall; thus converting the potential energy of the water into work in two stages. In the same way, the conversion of potential energy into work may be divided into three, four, or more stages. In the heat engine, the range of temperature through which the working fluid can fall is similar to the head of water; and this range of temperature may be divided into two, three or four stages, as may seem best. When so divided, we have compound, triple, and quadruple-expansion engines. These terms apply to the number of stages employed—not to the number of cylinders.

While, in the steam engine, the reduction of temperature of the steam used is the object striven for, yet we are accustomed to deal with the temperature only indirectly. If the steam is saturated, a given temperature of steam will be accompanied by a certain pressure, so that when we reduce the pressure of the steam as low as possible, we are also reducing the temperature. In many ways it is more convenient to deal with the pressure of the steam, but the connection between pressure and temperature should not be forgotten.

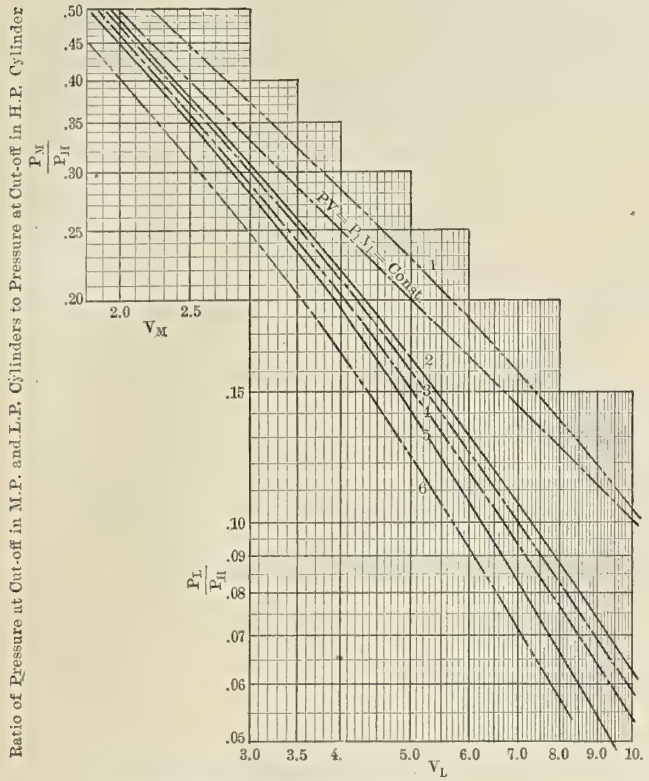
A reduction in the pressure of a gas may be obtained by causing it to occupy a larger volume, and while saturated steam is not a perfect gas, yet we can reduce its pressure by causing it to occupy successively larger volumes. The steam, in expanding, will exert against the piston of the cylinder in which the expansion is effected a constantly decreasing pressure, and this pressure can be used for doing external work. If the pressure against the piston were constant, the work done would be the product of the total pressure upon the piston and the distance through which it travels; but as the pressure is constantly decreasing, we must obtain in some way the average or mean pressure acting, and multiply that by the distance moved, the result being work done over that distance.

If the relation between pressure and volume is one that can be expressed mathematically, such as, for instance, $P_1V_1 = PV$, we can obtain the average pressure quite easily; but, unfortunately, the relation between the pressure of the steam and the volume occupied by it as it passes through the engine cannot be expressed in any such simple way. In Fig. 1 are shown some curves giving the relation between steam pressures and volumes, obtained from the indicator cards of numerous engines. The volumes used are the total volumes in the cylinders up to cut-off, including clearance, and the pressures are the absolute pressures at cut-off. It can easily be seen that no two engines would necessarily give the same curve of relation of pressures and volumes.

Suppose that we have two engines with high-pressure cylinders of the same size, taking steam at the same pressure, and the same pressure and volume in each at cut-off. Suppose that the same work is done in each, and the mean back pressures are the same in each. The pressures at cut-off in the next, or medium-pressure, cylinders will depend upon what happens to the steam in passing from one cylinder to the other. If the relative location of cylinders is different, so that the steam has to pass through longer pipes in one case than in the other; if the steam speeds used are different; if one pipe is better protected from radiation than the other;—the pressures at cut-off will be different in the next cylinders; so we cannot obtain any one curve that will give the relation between pressure and volume for all engines.

The curves 2, 3, 4 and 5, given in Fig. 1, are for the type of engine usual upon the ships of the merchant marine. The engine of a pumping plant, where every effort is made to get an economical engine by using jackets, reheaters, etc., will give a curve like 1; while the engine of a naval vessel may lie in the region of curve 6. These curves give the relation between pressure and volume ratios; the unit pressure and volume being that at cut-off in the high-pressure cylinder. In using such as curve for engine design it is best, wherever possible, to derive the curve from an engine similar to the one

* For the sake of simplicity the title of this article is MARINE ENGINE DESIGN, but it is desired to direct attention as well to the desirability of systematic data-keeping. Whenever possible, coefficients and factors of safety should be determined from data of similar engines working under as nearly as possible the same conditions. In this way only can it be claimed for an engine that it is *designed* for its work. Only the main points of marine engine design will be taken up, but the proportions of other parts can be determined in a similar manner.



Ratio of Vol's. at Cut-off (including Clearance) in M.P. and L.P. Cylinders to Vol. at Cut-off in H.P. Cylinder

FIG. 1.

projected, but if no such data are at hand, a curve can be chosen from the region of curves 2, 3, 4 and 5, in Fig. 1. Curve 3 is for a quadruple expansion engine in which the steam speeds varied from 7,000 to 11,000 feet per minute. Curve 4 is for a naval engine, whose data are given in Cards I. and II. The steam speeds here varied from 8,000 to 13,000 feet per minute. The more efficient engines with the lower steam speeds will probably lie in the neighborhood of curves 2 and 3, while higher steam speeds and a lower efficiency will cause them to lie in the neighborhood of curves 4 and 5. The way in which such curves can be used for purposes of design will be explained later on.

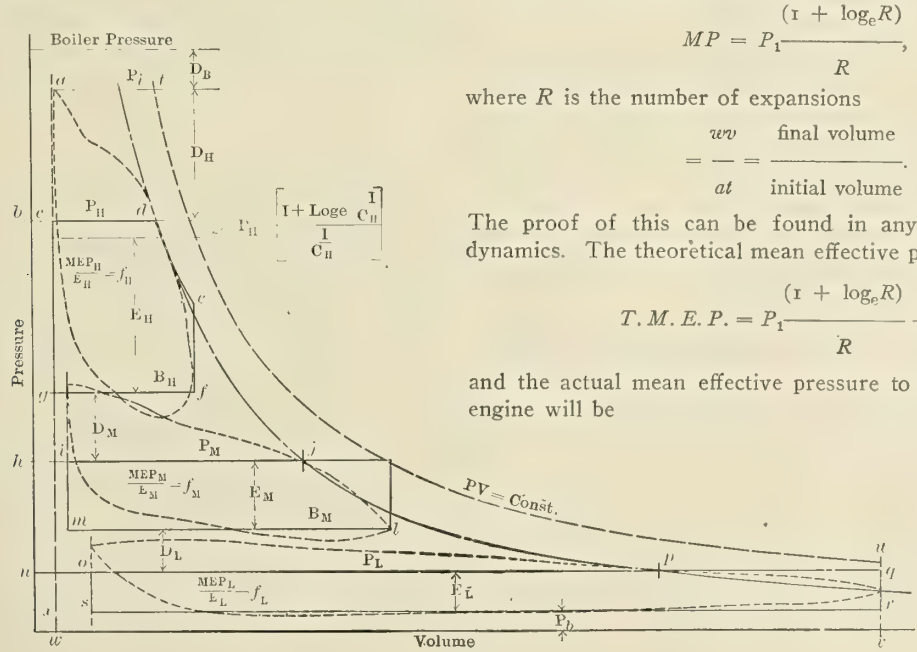


FIG. 2.

CYLINDER DIMENSIONS.

There are certain data which must be available before a start can be made upon the design of any engine. It is usual to give the desired indicated horsepower, the boiler pressure, the piston speed and the number of expansions. The piston speeds for engines of the merchant marine usually range from 600 feet per minute to 1,000 feet per minute. The higher the piston speed the more attention the engine will require; the greater will be the bill for repairs, and the shorter will be the engine's life. The number of expansions to be used for a given boiler pressure will be determined by experience, and will be a compromise between the number which will give the greatest economy and that which will give the engine of least weight.

The formula for indicated horsepower is

$$I.H.P. = \frac{2plan}{33{,}000}, \qquad (1)$$

where p is the mean unit pressure acting upon an area a, through a distance of $2ln$, or the piston speed; l is the length of stroke in feet, and n is the revolutions per minute. Referring to the data supposed to be available, it will be seen that the unknowns are

$$pa = \frac{I.H.P. \times 33{,}000}{\text{piston speed}} \qquad (2)$$

As stated before, work is derived from the steam by reducing its temperature by means of expansion; the increase in volume and decrease in temperature being accompanied by a decrease in pressure. The actual rate at which the pressure decreases cannot be stated in any simple form, but it is usual and convenient to assume the relation between pressure and volume to be given by the formula $P_1 V_1 = PV =$ constant; the character of the curve being that shown in Figs. 1 and 2. The deviation of the actual rate from the assumed rate, and certain losses, are allowed for by using a design factor derived from engines in use. The design factor F for the engine whose data are given in Cards I. and II., is 0.577, and is obtained by dividing the area inclosed in the dotted lines of Fig. 2 by the area $aturx$.

The mean pressure exerted upon an unit area while the steam is expanding from an initial pressure P_1 and volume at, to the volume wv, will be given by the formula

$$MP = P_1 \frac{(1 + \log_e R)}{R}, \qquad (3)$$

where R is the number of expansions

$$= \frac{wv}{at} = \frac{\text{final volume}}{\text{initial volume}}.$$

The proof of this can be found in any book on thermodynamics. The theoretical mean effective pressure will be

$$T.M.E.P. = P_1 \frac{(1 + \log_e R)}{R} - P_b, \qquad (4)$$

and the actual mean effective pressure to be expected in the engine will be

$$MEP = \left[P_1 \frac{(1 + \log_e R)}{R} - P_b \right] F. \qquad (5)$$

This MEP in (5) will be the p of (2), so that the value of the only remaining unknown a can be found. This a will be the area of the cylinder of the single-stage expansion engine, or the area of the low-pressure cylinder of any engine where the expansion or reduction of temperature is effected in more than one stage. In order to see this, it is only necessary to remember that in order to produce the desired indicated horsepower, starting with steam of a given pressure and expanding it a certain number of times, a certain weight or volume of steam will have to be used. After this steam has been expanded the chosen number of times, it will occupy a definite volume, irrespective of the number of stages of expansion, and that final volume will be the volume of the low-pressure cylinder. Therefore,

$$LP = a = \frac{I.H.P. \times 33{,}000}{P.S. \times MEP}. \qquad (6)$$

A low-pressure cylinder having been provided, of sufficient capacity to allow for the desired expansion of the weight of steam necessary to produce the desired indicated horsepower, the next step is to provide a cylinder of sufficient size to take in the necessary amount of steam from the boiler. As it is desirable that some portion of the total expansion should take place in this cylinder, it will not take steam from the boiler during its full stroke, but will cut off communication at some fraction of the stroke. The number of expansions

$$R = \frac{\text{final volume of steam}}{\text{initial volume of steam}} =$$

$$\frac{\text{area of L. P. cyl.} \times \text{stroke}}{\text{area of H. P. cyl.} \times \text{stroke} \times \text{fraction of cut-off}} = \frac{LP}{HP \times C_h}.$$

Therefore,

$$HP = \frac{LP}{R \times C_h}. \qquad (7)$$

If the expansion takes place in one cylinder, or in a single-stage engine, then

$$HP = LP, \text{ and } C_h = \frac{LP}{HP \times R} = \frac{1}{R};$$

and the cut-off fraction will be the reciprocal of the number of expansions.

Between the high-pressure cylinder and the low-pressure cylinder we can put as many cylinders as is thought advisable. The number will not affect materially the total power developed, but will merely divide the expansion into as many stages as is necessary for economical use of the steam. The sizes of these intermediate cylinders, and the cut-offs in all cylinders except the high-pressure cylinder, will affect only the distribution of work among the cylinders.

The reason for using two or more stages for the expansion of the steam is to avoid the loss by initial condensation that occurs when the range of temperature in one cylinder is more than 75 or 80 degrees F. In a double-acting engine, at the beginning of the stroke, there is upon one side of the piston, steam at the temperature of the source of supply, and on the other, steam at the temperature of exhaust. As the piston moves, the high-temperature steam comes in contact with a portion of the cylinder wall which just previously has been in contact with the low-temperature steam, and condensation results. It is to reduce this difference of temperature, and the consequent condensation, that the range of temperature in one cylinder is limited by dividing the total expansion into two or more stages. Thus the number of stages will depend upon the total range of temperature, or upon the initial pressure of the steam to be used. In Table I. are shown the values of the different quantities for merchant engines and naval engines.

TABLE I.

Type.	Initial Pressure at Engine, Absolute. P	Nominal Expansions. R	High-pressure Cutoff. C_h	Back Pressure in Low-pressure Cylinder. P_b	Design Factor. F
Triple naval.....	165 to 200	6 to 8½	0.70 to 0.80	5 to 7	0.58 to 0.70
Triple merchant.....	165 to 185	9 to 12	0.65 to 0.75	5 to 6	0.65 to 0.75
Quadruple merchant...	190 to 210	12 to 14	0.65 to 0.75	5 to 6	0.55 to 0.70

Collecting together the principal formulæ used so far, we have:

$$MEP = \left[P_1 \frac{(1 + \log_e R)}{R} - P_b \right] F \qquad (5)$$

$$\text{Area of } LP = \frac{I.H.P. \times 33{,}000}{PS \times MEP} \qquad (6)$$

CARD I.

Name of Ship.	H.P.		1 M.P.		2 M.P.		L.P.	
Size of engine 27—41—64 / 30	top	bot.	top	bot.	top	bot.	top	bot.
Cutoff—Designed or apparent,	.77	.71	.78	.68			.75	.69
Cutoff—Mean,	.74		.73				.72	
Clearances,								
Clearances—Mean of top and bottom,	.134		.10				.069	
Initial pressure—Absolute,	163	163	78	76			27	29
Cutoff pressure—Absolute,	123	127	49	54			17¼	19¾
Cutoff pressure—Mean absolute,	125		51.5				18.5	
Average back pressure—absolute,	73		31				6.5	
Effective cutoff pressure,			20.5				12.0	
M. E. P. from indicator cards,	53.5	59.2	23.5	25.4			14.0	13.4
Mean of top and bottom, M. E. P.,	56.4		24.4				13.7	
I. H. P. developed in cyl. P. S. = 838	824		820				1115	
M. E. P. when power is equally distrib.,	63.3		27.4				11.26	

$$\text{Area of } HP = \frac{LP}{R \times C_h} \qquad (7)$$

The stroke has usually some one of the following values: 18, 21, 24, 27, 30, 33, 36, 39, 42, 45, 48, 54, 60, 66, or 72 inches. The ordinary merchant engine will have one of the above values of stroke, nearest the square root of the product of the diameters of the high-pressure and low-pressure cylinders. In naval engines the stroke will be less than this, in extreme cases being only 50 percent of the above value. On the other hand, in slow-running, economical freight steamer engines, it will be greater by 10 to 15 percent.

The proper diameter for the intermediate-pressure cylinder or cylinders, and the cut-off in the mean-pressure and low-pressure cylinders, will be chosen to give a good distribution of work among the cylinders. In order to make this choice with intelligence, the data obtained from engines in service should be worked up in such a way as to be readily used. A system of data keeping which has been found to give good results deals with the following quantities and relations: The curve of relation between cut-off pressures and total volumes

at cut-off in the different cylinders (see curve *djp*, Fig. 2); the drop D_h from the initial pressure to the cut-off pressure in the high-pressure cylinder; the mean back pressure in each cylinder, B_h, B_m, P_b; the drop from the mean back pressure of one cylinder to the cut-off pressure of the next succeeding, D_m and D_l; the relation between the mean effective pressure of the high-pressure indicator card MEP_h and the mean effective pressure E_h of a card *cdefg*, which assumes that the steam enters at the cut-off pressure and expands with PV = constant,

$$\frac{MEP_h}{E_h} = f_h;$$

the relation between the MEP of the mean-pressure and low-pressure cylinder indicator cards and the effective cut-off pressures in those cylinders,

$$\frac{MEP_m}{E_m} = f_m, \quad \frac{MEP_l}{E_l} = f_l.$$

This data can be kept on a card similar to that shown in Card I., and when worked up, as shown in Card II. on the back of the card, the necessary factors and relations will be available.

There are various ways of getting the design factor F. In this system of data keeping the engine is not charged with the loss of pressure that occurs between the boiler and the engine,

CARD II.

No. of expansions $= \left(\frac{64}{27}\right)^2 \frac{1}{0.74} = 7.61$

$163 \frac{3.0295}{7.61} - 6.5 = 65 - 6.5 = 58.5$

$56.4\left(\frac{27}{64}\right)^2 = 10.05$

$24.4\left(\frac{41}{64}\right)^2 = 10.02$

13.7

33.77

$\frac{24.4}{20.5} = 1.19 = f_m$

$\frac{13.7}{12} = 1.14 = f_l$

For power equally distributed

$\frac{33.77}{3}\left(\frac{64}{27}\right)^2 = \frac{63.3}{56.4} \quad \frac{63.3}{47+6.9} = 1.17 = f_h$

6.9

$\frac{33.77}{3} \times \left(\frac{64}{41}\right)^2 = \frac{27.4}{24.4} \quad \frac{27.4}{20.5+3} = 1.165 = f_m$

$3.$

$\frac{33.77}{58.5} = 0.577 =$ Design factor F.

$D_h = 163 - 125 = 38$
$D_m = 73 - 51.5 = 21.5$
$D_l = 31 - 18.5 = 12.5$

$11.26 - 13.7 = -2.4$

$\frac{11.26}{12-2.4} = 1.172 = f_l$

$\frac{72}{163} = 0.442$ $\qquad 72.0 = D$

$\frac{51.5}{125.} = 0.412 = P_m$ $\qquad \frac{18.5}{125.} = 0.148 = P_l$

$125 \frac{1 + 0.3001}{1.35} = 120 \quad \frac{73}{47}$

$\frac{56.4}{47} = 1.20 = f_h$

$\frac{(41)^2 \times (0.73 + 0.10)}{(27)^2 \times (0.74 + 0.134)} = 2.19 = V_m$

$\frac{(64)^2 \times (0.72 + 0.069)}{(27)^2 \times (0.74 + 0.134)} = 5.07 = V_l$

as this will vary with the relative location of engine and boilers, steam speeds used and general layout of piping. The values given for F, in Table I., consider the steam as entering the engine with the initial pressure shown on the high-pressure indicator card, expanding the nominal number of times with $P_1V_1 = PV$ = constant, and with a back pressure equal to the mean back pressure of the low-pressure cylinder indicator card. The nominal number of expansions does not take into consideration the reduction of expansion due to clearances.

The design factor F takes account not only of the deviation of the actual relation of pressure and volume from the assumed, but also of the losses of pressure that occur while the steam is passing from one cylinder to the next. We should expect, therefore, that the sum of the drop $D = D_h + D_m + D_l$ would depend upon the value of the design factor. It has been found that there is such a relation, and the approximate values of D are given in connection with F, as a portion of the absolute initial pressure P_1.

TABLE II.

F =	0.6	0.65	0.675	0.70	0.725	0.75	0.80
$\frac{D}{P_1}$ =	.35—.40	.25—.29	.235—.27	.225—.25	.21—.23	.20—.215	.175—.185

$D_h = 0.5$ to $0.6D$; $D_m = 0.25$ to $0.35D$; $D_l = 0.12$ to $0.17D$

The drop D_b from boiler to engine will vary from 10 to 25 pounds, usually being from 15 to 20 pounds. The relative values of D_h, D_m and D_l will not be constant, as can be easily seen when the causes of the drop are considered. No two engines will necessarily work under the same conditions, so that the relative values can be only approximately indicated. Whenever possible, the various factors, drops, etc., should be derived from engines similar to the one projected, working under like conditions.

The card factors f_h, f_m and f_l should be reduced to some standard condition. The character of the admission line and back pressure line of an indicator card will not change essentially if there is 10 percent more or 10 percent less work developed in the cylinder. The two lines will be bodily separated or brought closer together. The standard condition chosen in the case of the triple engine is that which exists when one-third of the total indicated horsepower is developed in each cylinder. Thus, if the MEP of an indicator card is 31, and the effective cut-off pressure is 28, while the MEP should be 33 in order that one-third of the work may be done in that cylinder, the corrected value of f_m is

$$\frac{31 + 2}{28 + 2} = 1.1.$$

It can be seen by referring to Fig. 2 that f_h will vary with the drop D_h, and f_m and f_l will vary with the fraction of cut-off in the cylinder. It was found that the values of f_h, f_m and f_l, when reduced to the standard condition, could be given by the formulæ,

$$f_h = 1 + \frac{D_h - 16}{100} \qquad (8)$$

$$f_m = 2.25\ C_m - 0.475 \qquad (9)$$

$$f_l = 1.25\ C_l + 0.256 \qquad (10)$$

where D_h is the drop from initial to cut-off pressure, and C_m and C_l are the fractional cut-offs in the mean-pressure and low-pressure cylinders. The way in which these factors and relations of pressure and volume can be used for design purposes can be best shown by means of an example:

I. H. P. = 3,000; *P. S.* = piston speed = 850 feet per minute; boiler pressure = 185 pounds gage; P_b = back pressure in low-pressure cylinder = 5.5 pounds; R = number of expansions = 11; C_h = cut-off in high-pressure cylinder = 0.675; F = design factor = 0.7. Work to be equally divided among the three cylinders.

The initial pressure at the engine will be taken as 170 pounds gage, or 185 pounds absolute, allowing a drop of 15 pounds from the boiler to the engine. Table II. gives the allowable total drop for a design factor of 0.7 as 0.225 to 0.25 of P_1. We will use $D = 0.24\ P_1 = 0.24 \times 185 = 44.5$ pounds.

$$D_h = 44.5 \times 0.55 = 24.5; \quad D_m = 44.5 \times 0.30 = 13.5;$$

$$D_l = 44.5 \times 0.15 = 6.5$$

From (5) $$MEP = \left[185 \frac{1 + 2.398}{11} - 5.5\right] \times 0.7$$

$$\left[= 57.1 - 5.5\right] \times 0.7 = 36.1$$

From (6) $$LP = \frac{3{,}000 \times 33{,}000}{850 \times 36.1} = 3{,}230$$ square inches; diameter = 64 inches.

From (7) $HP = \dfrac{3{,}230}{11 \times 0.675} = 435$ square inches; diameter $= 23\frac{1}{2}$ inches.

$\sqrt{64 \times 23\frac{1}{2}} = 38.8$. Let stroke = 42 inches.

In order that one-third of the total work may be done in the high-pressure cylinder, the *MEP* of the indicator card must be

$$MEP_h = \frac{36.1}{3} \times \left(\frac{64}{23.5} \right)^2 = 89 \text{ pounds.}$$

From (8) $f_h = 1 + \dfrac{24.5 - 16}{100} = 1.085$; $E_h = \dfrac{89}{1.085} = 82$ pounds

$P_h = 185 - 24.5 = 160.5$ pounds.

$$B_h = P_h \frac{1 + \log_e \frac{1}{C_h}}{\frac{1}{C_h}} - E_h = 160.5 \times \frac{1 + 0.392}{1.48} - 82 = 152 - 82 = 70$$

$P_m = B_h - D_m = 70 - 13.5 = 56.5$ pounds.

$\dfrac{P_m}{P_h} = \dfrac{56.5}{160.5} = 0.352.$

The volume ratio that will give this pressure ratio, 0.352, will be found by using curve 3 in Fig. 1. The volume ratio $V_m = 2.63$. These values of P_m and V_m will be used later. Having found P_m, we will now proceed to get B_m by working up from the low-pressure cylinder.

We have assumed that curve 3, Fig. 1, gives the relation between pressure ratios and volume ratios for the engine we are designing, so that when we assume a certain cut-off in the low-pressure cylinder the cut-off pressure is determined from the curve. This cut-off pressure must be sufficient to give an effective cut-off pressure E_1, of such a value as will insure that one-third of the work is done in that cylinder. The relation between E_1 and MEP_1 will be given by (10).

In order to get the volumes up to cut-off, we must know the clearances. It will be best to determine these by comparison with a similar engine, but if no such data are at hand the following clearances can be assumed for merchant ship engines, where the stroke $= \sqrt{\text{high-pressure diam.} \times \text{low-pressure diam.}}$:

High-pressure cylinder	0.17 to 0.20.
Medium-pressure cylinder	0.13 (piston valves); 0.11 (slide valves).
Low-pressure cylinder	0.12 (piston valves); 0.09 (slide valves).

These values will increase as the stroke gets relatively shorter, so that they should be mutiplied by

$$\sqrt{\frac{\text{High-pressure diameter} \times \text{low-pressure diameter}}{\text{stroke}}}$$

to get the approximate clearances in other cases.

We will take the clearances as $Cl_h = 0.17$, $Cl_m = 0.13$, $Cl_l = 0.12$ for the high-pressure, mean-pressure and low-pressure cylinders, respectively,

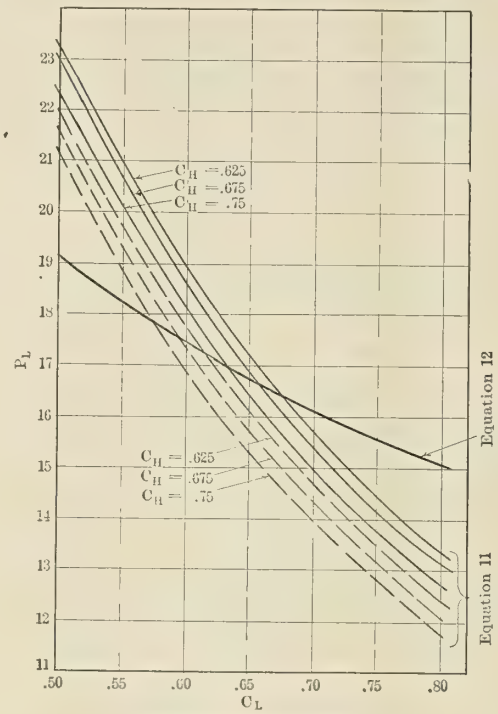
The value of P_1 can be obtained from curve 3, Fig. 1, as follows: The volume ratio will be

$$V_1 = \frac{(C_1 + Cl_1)LP \times s}{(C_h + Cl_h)HP \times s}. \qquad (11)$$

With this value of V_1 the value of P_1/P_h can be obtained from the curve, and knowing P_h, the value of P_1 is known. The value P_1 must also be such as will cause the necessary work to be done in the cylinder, or,

$$P_1 = P_b + \frac{MEP_1}{1.25C_1 + 0.256} \qquad (12)$$

In order to find the value of P_1 which will satisfy both equations (11) and (12), we will assume values of C_1, plot the resulting values of P_1 on C_1, and find where the two curves cross.



Full Lines from Curve 3, Fig. 1.
Broken " " " 4, Fig. 1.

FIG. 3.

From (11) $V_1 = \dfrac{(C_1 + 0.12)3{,}230}{(0.675 + 0.17)435} = (C_1 + 0.12) \times 8.8$

From (12) $P = 5.5 + \dfrac{12.03}{1.25\,C_1 + 0.256}$

TABLE III.

C_1	V_1	From Fig. I. P_1/P_h	P_1
0.5	5.45	0.144	23.1
0.6	6.33	0.116	18.6
0.7	7.22	0.096	15.4
0.8	8.08	0.082	13.2

C_1	P_1
0.5	19.17
0.6	17.46
0.7	16.17
0.8	15.1

When these values are plotted upon C_1 as abscissæ, the resulting curves cross at a pressure of 16.65 pounds, and with $C_1 = 0.655$ (see Fig. 3), $C_h = 0.675$.

$$B_m = P_1 + D_1 = 16.65 + 6.5 = 23.15$$

The value of $P_m = 56.5$ has been previously obtained.

$$E_m = P_m - B_m = 56.5 - 23.15 = 33.35$$

We are now ready to determine the mean-pressure diameter and cut-off. Here again there are two conditions which must be fulfilled, and consequently two equations to use. The value of E_m, when multiplied by the factor f_m, depending upon the cut-off, must give a MEP_m that will do the necessary work when acting upon the mean-pressure cylinder area. This gives the equation:

$$E_m \times f_m = MEP_m = \frac{MEP \times LP}{3 \times MP} = E_m(2.25C_m - 0.475)$$

$$\text{or} \qquad MP = \frac{MEP \times LP}{3E_m(2.25C_m - 0.475)}. \qquad (13)$$

This area of mean-pressure cylinder and cut-off must also be such as will give the proper volume ratio V_m previously obtained from the pressure ratio P_m/P_h.

$$\frac{MP(C_m + Cl_m) \times s}{HP(C_h + Cl_h) \times s} = V_m$$

$$\text{or,} \qquad MP = \frac{V_m \times HP(C_h + Cl_h)}{C_m + Cl_m} \qquad (14)$$

We can equate (14) and (13) and solve for C_m, the only unknown.

Let $X = V_m \times HP(C_h + Cl_h)3E_m$, and
$Y = MEP \times LP$.

$$\text{Then,} \qquad CM = \frac{0.475X + Cl_m Y}{2.25X - Y}.$$

This value of C_m, substituted in either (13) or (14), will give the area of the mean-pressure cylinder. For the engine we are designing.

$$X = 2.63 \times 435(0.675 + 0.17) \times 3 \times 33.35 = 96{,}200.$$
$$Y = 36.1 \times 3{,}230 = 116{,}700.$$
$$C_m = \frac{0.475 \times 96{,}200 + 0.13 \times 116{,}700}{2.25 \times 96{,}200 - 116{,}700}$$
$$= \frac{45{,}700 + 15{,}150}{217{,}000 - 116{,}700} = \frac{60{,}050}{101{,}300} = 0.60$$
$$MP = \frac{2.63 \times 435(0.675 + 0.17)}{0.60 + 0.13} = 1{,}328, \text{ diameter} = 41 \text{ inches.}$$

Collecting together the quantities calculated and assumed so far, we have

	High-Pressure.	Medium-Pressure.	Low-Pressure.
Diameter in inches	23½	41	64
Cut-off	0.675	0.60	0.655
Clearance	0.17	0.13	0.12
Stroke, inches	42	42	42

If the high-pressure cut-off had been taken as 0.625, the results would have come as follows (see Fig. 3, full line curve, $C_h = 0.625$):

	High-Pressure.	Medium-Pressure.	Low-Pressure.
Diameter in inches	24½	42	64
Cut-off	0.625	0.56	0.67

If the high-pressure cut-off had been taken as 0.75, the following results would have been obtained (see Fig. 3, full line curve, $C_h = 0.75$):

	High-Pressure.	Medium-Pressure.	Low-Pressure.
Diameter in inches	22½	39½	64
Cut-off	0.75	0.71	0.635

In computing the foregoing results, curve 3 in Fig. 1 was used. If curve 4 had been used, the results would have been as follows (see Fig. 3, broken curves):

	High-Pressure.	Medium-Pressure.	Low-Pressure.
Diameter in inches	23½	39	64
Cut-off	0.675	0.655	0.595
Diameter in inches	24½	40	64
Cut-off	0.625	0.61	0.615
Diameter in inches	22½	36¾	64
Cut-off	0.75	0.81	0.575

It may not be thought advisable to use any one of the sets of results as it stands. The effect of any change, however, can be foreseen, as an increase of the cut-off in any cylinder will cause less power to be developed there and more power in the preceding cylinder. A shortening of the cut-off will have the reverse effect. It will be noticed that an increase of the high-pressure cut-off is accompanied by an increase of the mean-pressure cut-off, and a decrease of the low-pressure cut-off if equal powers are to be developed in each cylinder; while a decrease in the high-pressure cut-off causes the mean pressure to decrease and the low pressure to increase.

(To be Continued.)

PRACTICAL EXPERIENCE WITH MARINE STEAM TURBINES.

Built, building, or on order, there are now some 140 vessels fitted with marine turbines, mostly of the Parsons type. Signs, however, are not lacking that other systems will soon be acknowledged competitors, the sound, practical features of the impulse type of machine giving it many advantages. Some of these vessels have now been in service for several years, and information is rapidly accumulating with regard to the once uncertain questions of durability, operation and cost of maintenance. So much has been written and said, on short acquaintance, of the virtues of marine turbines that their defects are seldom heard of, but experience proves that the annual overhaul of the machinery occasionally involves a good deal of work. In many cases, however, the difficulties involved have led to improved design of the details affected, or to more careful methods of operation.

One of the most prolific causes of trouble is the use of dirty feed-water. Elaborate strainers are fitted on each steam pipe to the main turbines to catch any solid matter coming over from the boilers, but the finer particles get through and frequently choke up the passages between the smaller rows of blades. The result is that a higher steam pressure (which is not always available) is necessary to maintain the same speed, and the trouble of removing the scale formed on the blades is considerable. This deposit is frequently found on the inlet blades of the low-pressure turbines, the surfaces of which become coated like the inside of a kettle. As long as this hard scale adheres to the blades its presence is detrimental only to economy, but in the event of pieces becoming detached, "blade stripping"—with which we propose to deal at length in a later issue— is extremely likely to occur. On the other hand, should the grit, caused by the disintegration of the scale, get into the "dummy packing," which depends on very fine clearances, the brass strips are likely to be badly worn, thereby increasing the clearances and the leakage loss.

Another effect of dirty feed-water is to induce priming, and while it is doubtful whether small quantities of water are sufficient to cause blade stripping, severe stresses on the blading are set up if much priming occurs. On their early voyages, both the *Victorian* and the *Virginian* seem to have suffered from this, and the effect of excessive scale has been noted on several of the channel steamers, as well as in large land installations. The warships belonging to the royal navy do not seem to be troubled, but in their case great care is taken to use only clean feed-water, and attention should always be paid to this point in turbine vessels.

It has been found that the main bearings, which in reciprocating engines constantly require attention, give practically no trouble whatever for months together, provided that the lubrication system is kept in good order. The pressure being low—generally only about 30 pounds per square inch—and the direction of rotation and pressure practically constant, the wear observed after twelve months' running is practically nil, and cases can be cited of marine turbine bearings that have run for over three years without adjustment.

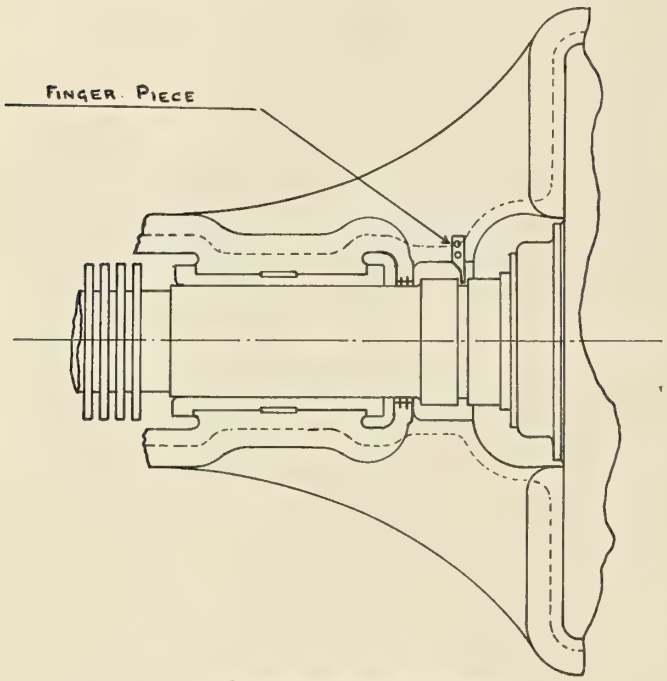


FIG. 1.—PLAN OF TURBINE BEARING.

The same applies to thrust blocks, in the case of which little attention is necessary when the turbine is at work. They perform the function, however, of an adjusting block as well, for the main turbines, and the rotor is set for dummy clearance by their means. This adjustment is a somewhat "ticklish" job. The rotor is so designed that the pressure, due to the steam acting on the blades, is rather greater than that due to the propeller; hence any wear that takes place in the thrust collars does so in the direction of the main steam flow; the rotor consequently tends to recede from the dummies. A groove in the shaft between the bearing and the gland, and a finger-piece set in a definite position on the cylinder and projecting into the groove, as shown in Fig. 1, provide a ready means of determining the amount of wear. At convenient intervals the shafting is disconnected aft of the rotor, and the latter is drawn up to the prescribed clearance again by means of the adjusting block screws. The wear, however, is so slight, that, as a rule, this adjustment requires to be made only once or twice a year.

Another feature that needed modification in a great many of the earlier vessels was the astern dummy ring. On account of the longitudinal expansion of the rotor, these dummy rings had to be made radial; they were placed on an extension of the astern turbine drum, which, having to carry only these rings, was made very thin. In numerous cases this drum became bell-mouthed when running, owing to centrifugal force, thereby wearing off the dummy rings and causing excessive leakage when running astern. This early design is shown in Fig. 2, while Fig. 3 gives the stiffer modern type now adopted with success. Fig. 4 shows the arrangement of astern dummy rings.

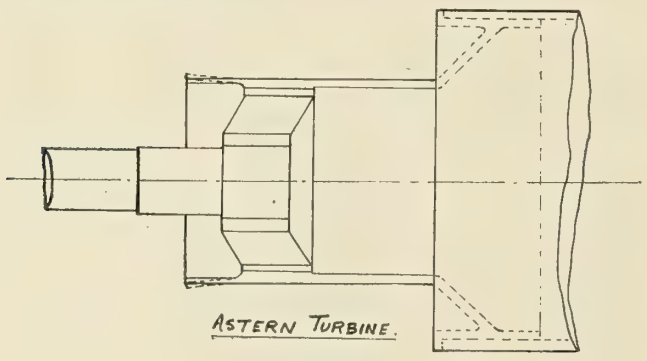


FIG. 2.

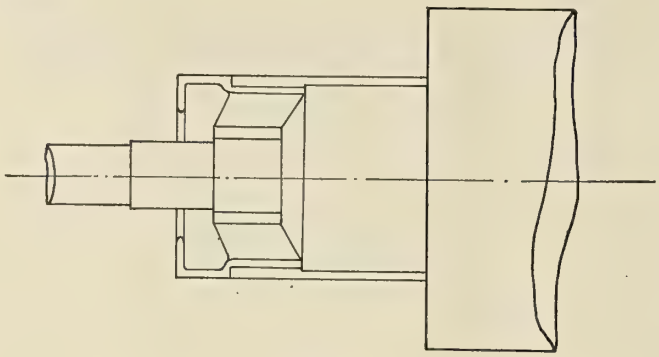


FIG. 3.

An utterly unsuspected source of considerable trouble in marine turbine plants has been found to lie in leaky condenser tubes, causing corrosion of the rotor drums of the low-pressure turbines. In some cases this phenomenon has proved extremely serious. Generally speaking, turbines run so satisfactorily in service that the covers seldom need to be lifted, and many cases can be quoted of vessels requiring this attention only once a year. Perhaps the best instance of this kind is the *Carmania*. Her turbines remained closed from the time they were put on board the ship until the completion of twelve months in service, and the ship has just now completed a second year's service, when they were lifted again. Such immunity involves trusting to luck, to some extent, as to the internal condition of the turbine. The pressure gages and tachometers show if any stripping or choking of the blades has occurred, but the out-of-balance that results from unequal corrosion is generally first put down to the propeller being damaged.

In the case of several cross channel steamers—particularly those running on the Great Western Railway Company's service between Fishguard and Rosslare—it was noticed that the rotors were out of balance, and at the annual overhaul abnormal corrosion was found to have attacked the insides of the low-pressure turbine drums, as well as all wrought iron

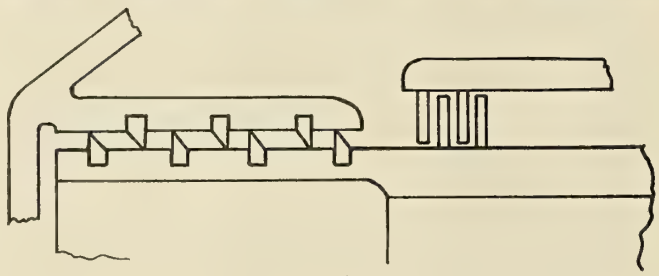


FIG. 4.

nuts inside the casing. This corrosion arises chiefly from salt water leaking back from defective condenser tubes through the non-return valves on the low-pressure drain pipes, and if neglected may prove very serious. The external portions of the rotor are not attacked, curiously enough, leading to the supposition that in addition to salt water, stagnant water that may have been unable to escape from the interior of the drum, owing to the choking up of the drainage grooves, has been whirled around and around inside, causing severe erosion.

A good deal of air probably leaks into the turbine through the glands. Those on the low-pressure turbines have to keep tight against atmosphere on one side and a high vacuum on the other, and in some cases, if the steam supply to the glands is not adjusted properly, a considerable amount of air may

get in. The water sealed gland so much preferred in American land turbine practice does not seem to be used in marine work. In any case, the prevention of salt water and air leakage into the turbine should go far to obviate any chance of corrosion. It is a defect that has not been found in well-kept engine rooms.

Glands often require a great deal of attention. In all the early turbine steamers they could be overhauled only by lifting cover and rotor, as they consisted of Ramsbottom rings fitted in grooves in the shaft. It was not for some years that Parsons adopted the external gland, which seemed so essential to convenience. These Ramsbottom rings wear very quickly sometimes, but this fact is easily noticed on account of the greatly increased flow of steam past them into the engine room. There is still considerable room for improvement in turbine glands; their essentials are shortness longitudinally, tightness without wear, and ease of adjustment, and these are not easy to arrive at.

Contrary to a popular idea, not only is vibration frequently met with on turbine steamers, but that vibration is sometimes excessive. The *Lusitania, Mauretania* and *Carmania* are by no means free from it, and some of the channel steamers are

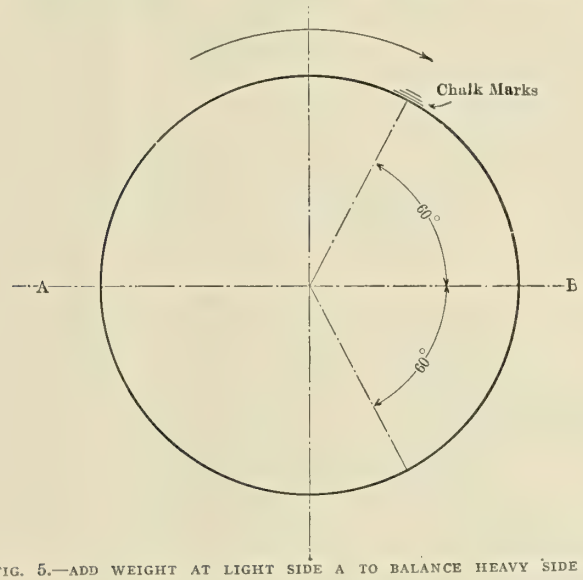


FIG. 5.—ADD WEIGHT AT LIGHT SIDE A TO BALANCE HEAVY SIDE B.

also bad. It is seldom or never due to the turbine, unless a bad strip or corrosion has taken place, but it is due entirely to the propeller. Either the screw is too near the stern post or the hull, or, as in the case of the *Dreadnought,* to the rudders, though in this latter case the rudders suffered more than anything else. The high speed of rotation and the varying intensity of the stream lines do not facilitate smooth working, if there is the slightest tendency for the screw not to get a proper supply of water. Turbine steamers need an exceptionally clean run of water to the propellers.

In balancing turbine rotors in the shop, two methods are generally used. First, the rotor is placed on knife edges and slowly moved, first one way and then the other, to determine the heavy side, if there is one. This can be corrected very accurately by this method, and it will generally be found best to cut off weight from the heavy side, rather than to add it to the light side. Very little is required to correct a rotor that is even badly out of balance, but with good workmanship the latter very seldom occurs. The second method is to run the rotor up to speed in the turbine under steam, and to determine the heavy side by chalking the shaft, which tends to revolve eccentrically. It should be noted, however, that the chalk marks on the shaft do not correspond exactly with the heavy side, but occur at a spot about 60 degrees behind it; this, of course, requires to be taken into account.

In the case of land turbine rotors, which are sometimes run up to speed in special bearings by means of an electric motor, it is easy to arrange to run the shaft, first in one direction and then in the other, so that a mean position can easily be found for the heavy side. The difficult case occurs, of course, when the rotor is out of balance at opposite sides at the two ends. When vibration in turbine steamers can be prevented to the extent that it has been in some vessels, it is annoying to find it so developed in some others. Care in design and workmanship is all that is needed to avoid it.

Great care must be taken that the oil remains in good condition. The consumption of this item is very small indeed, as it is used over and over again, the system, consisting of oil tank, cooler and pump, being a closed one. The temperature at which the turbine bearings work is generally between 90° F. and 120° F., depending rather on their position—for instance, the forward high-pressure bearing, which is very near the steam belt, is obviously hotter than the after low-pressure one. A cooler is essential, and the oil, after leaving the bearings, passes through a spiral coil, which is immersed in circulating water. This coil needs to be a good tight job, as cases have been known where water has leaked into the oil supply, with disastrous results. Ample cooling surface should be allowed.

Marine turbines seldom or never work with superheated steam, as is the case with land turbines. In the latter, the bearing temperatures are often very high, cases up to 190° F. having been known to run without trouble. Special oils are necessary for such conditions, but for ordinary work "Valvoline" has been found to give excellent results. It is a great mistake to use too high an oil pressure—5 to 6 pounds per square inch is ample, and the supply should be maintained very regular. A gravity system, in which the oil supply is situated high up in the engine room hatch, is very useful; the pump is then used merely to force the oil through the coolers and up to the tank.

SPEED TRIALS AND SERVICE PERFORMANCE OF THE CUNARD TURBINE STEAMER LUSITANIA.*

BY THOMAS BELL.

It is with feelings of some diffidence that I place before the members of this institution a brief record of the trials and running in service of the Cunard turbine steamer *Lusitania,* for, apart altogether from the fact that the propelling machinery consists of Parsons turbines, the leading proportions of the design of the ship and machinery are the outcome of most careful deliberations on the part of the technical staff of the Cunard Company, of the turbine committee, of the Hon. C. A. Parsons and his assistants, of the Admiralty, of Lloyd's committee, and of the Board of Trade, in conjunction with the staff of the three firms intrusted with the design and construction of the two express Cunarders, so many of whom are distinguished members of this institution. Very full reports of the trials have been published in the technical press, but your council considered that the subject had not yet been exhausted, and it is hoped that this brief paper may furnish material for an interesting discussion on some points connected with this great Cunard enterprise.

As already described so fully in the technical and other journals, and as shown on accompanying diagrammatic plan, Fig. 1, the turbine machinery of the *Lusitania* consists of two pairs of compound turbines, a pair consisting of one high-pressure and one low-pressure unit, each of which actuates a line of shafting, so that there are four lines in all. The high-pressure turbines drive the wing shafts, and the low-pressure

* Read before the Institution of Naval Architects, April 9, 1908.

turbines the center shafts. At the forward end of the low-pressure turbines are placed the astern turbines. This particular arrangement, in which the two center propellers are used for maneuvering purposes, was forced on the designers by exigencies of space as the only possible one, and has proved most satisfactory.

The boilers, as will be seen from Fig. 2, are divided into four equal groups in separate watertight compartments. The coal bunkers are situated only at the sides in the three after boiler rooms, but in the forward boiler room, owing to the fineness of the ship, the capacity of the side bunkers is comparatively small, and a large athwartship bunker becomes necessary, in addition to those at the sides.

The following are the principal dimensions and particulars of the turbines, condensers, shafting and boilers:

Total number of furnaces..... = 192
Total grate surface..... = 4,048 square feet.
Total heating surface..... = 158,352 square feet.
Total length of boiler rooms..... = 336 feet.
Total length of main and auxiliary engine rooms..... = 149 feet 8 inches.

With regard to the running of this large machinery and boiler installation, it may be of interest to members to give detailed particulars of the forced lubrication system, and of the system of feed heating and supply to the boilers. Each of these systems demanded that adequate provision be made to meet all possible contingencies, and thus enable the engineers in charge to be free to attend to the numberless duties connected with the supply of electric power and lighting, the supply of hot and cold, fresh and salt water throughout the ship, the pumping of the bilge and of the ballast compartments, and last, but not least, the supervision of the large army of stokers or firemen and trimmers, and the regulation and distribution of the coal supply from the various bunkers.

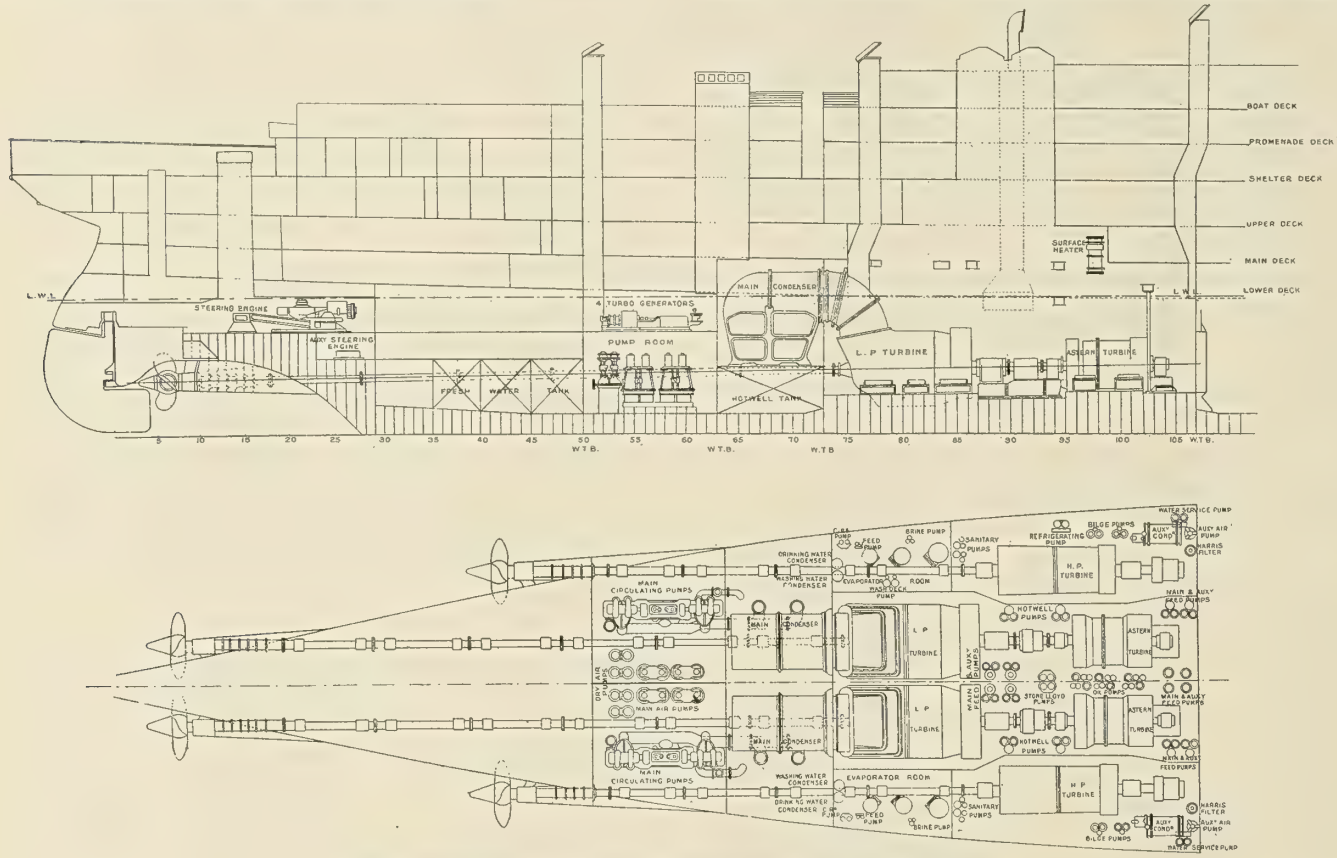


FIG. 1.—ARRANGEMENT OF TURBINES AND SHAFTING, STEAMSHIP LUSITANIA.

Turbines.	Diameter of Rotor.	Length of Blades. In First Expansion.	Length of Blades. In Last Expansion.
High-pressure	96 inches	2¾ inches	12⅜ inches
Low-pressure	140 inches	8¼ inches	22 inches
Astern	104 inches	2¼ inches	8 inches

Total cooling surface, main condensers..... = 82,800 square feet.
Area of exhaust inlet..... = 158 square feet.
Bore of circulating discharge pipes..... = 32 inches.
Diameter of tunnel shafts... = 20 inches external, 10 inch hole.
Diameter of propeller shafts = 22 inches external, 10 inch hole.
Boilers, working pressure, 195 pounds per square inch.
23 double-ended boilers, 17 feet 6 inches mean diameter by 22 feet long.
2 single-ended boilers, 17 feet 6 inches mean diameter by 11 feet 4 inches long.

Regarding the forced lubrication system, of which a diagrammatic sketch is given in Fig. 3, the following statement gives the weights of the various revolving parts, together with the size of bearings and the pressure on same:

Weight of one high-pressure turbine rotor complete.. = 86 tons.
Weight of one low-pressure turbine rotor complete... = 120 tons.
Weight of one astern turbine rotor complete..... = 62 tons.

	Main Bearing Journals. Diameter.	Main Bearing Journals. Effective Length.	Pressure Per Square Inch of Bearing Surface.	At 190 Revolutions, Surface Speed of Journal.
High-pressure rotor..	27⅛ inches	44⅜ inches	80 pounds	1,350 feet per minute
Low-pressure rotor..	33⅜ inches	56½ inches	72 pounds	1,650 feet per minute
Astern rotor.....	24⅛ inches	34¼ inches	83 pounds	1,200 feet per minute

A large main drain or reservoir tank is placed amidships, low down in the engine room, in such position that the oil from all the bearings readily flows into it; the oil is pumped from this tank by four direct-acting single-cylinder pumps, through a system of filters, to gravitation tanks placed in the engine room casings, about 26 feet above the bearings. These tanks are of sufficient capacity to maintain the oil supply to the bearings for ten or eleven minutes after the complete stoppage of all the pumps. There is a separate and distinct system of supply and gravitation pipes to the bearings on each side of the ship, with cross connections, so that, in the event of any defect arising on one side, it can be at once shut off, and the whole supply to all the turbines directed through the other side. Gages showing the pressure in the oil pump discharges to the filters, and also the pressure in the gravitation pipes to the main bearing journals and thrusts, are fitted up at the starting platform as well as at the pumps, these latter being placed in an easily accessible part of the center engine room. A pressure gage is also fitted at each bearing, and, in addition, the drain

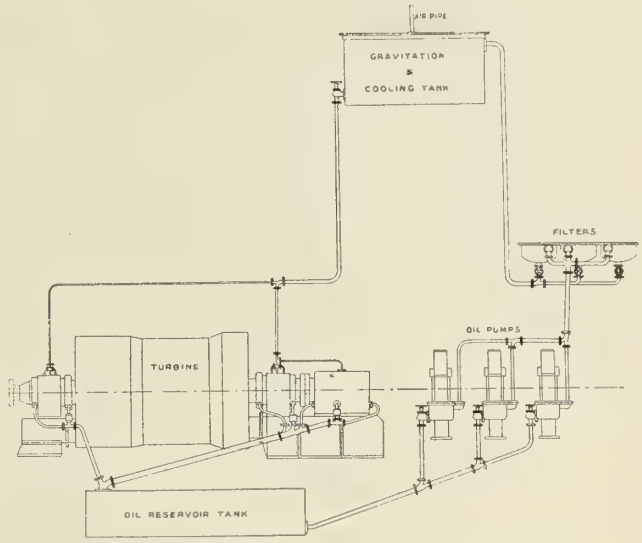


FIG. 3.—DIAGRAM OF FORCED LUBRICATION SYSTEM.

or discharge from the bottom reservoir from each bearing and thrust is led through a glass-sided lantern-shaped receiver, so as to make the flow visible, and a thermometer fitted up at this point enables the temperature of the oil from each of the twelve main bearings and four thrust blocks to be taken and recorded, the practice being to log the temperature hourly. In the gravitation tanks are placed copper coils of a total surface of 1,335 square feet, through which cold sea water is circulated, and this surface suffices to maintain, at a temperature not exceeding 30 degrees above that of the engine room, the 4,700 gallons of oil in circulation. Each coil is self-contained and withdrawable, so that a leakage of sea water from pitting or other cause can readily be located and repaired without interfering with the general working of the oil supply. There are also reserve oil tanks having a total capacity of 4,200 gallons for use in case of emergency.

Three of the six oil pumps, the discharges from which are all cross-connected, are ample for all the necessary oil supply, but four are kept in use, so that no risk is run in the event of the failure of any single pump not being at once noticed.

With reference to the feed system, shown diagrammatically in Fig. 4, it should be explained that all the auxiliaries in the ship, excepting the turbo-generators and the evaporators, exhaust into a general system of auxiliary exhaust piping connected to the surface feed heaters, auxiliary condensers and

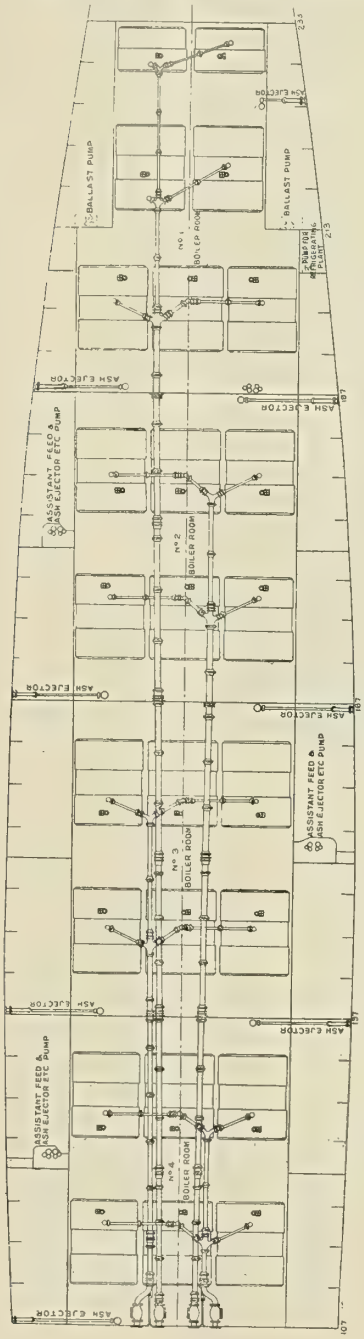


FIG. 2.—ARRANGEMENT OF BOILER ROOMS IN LUSITANIA, SHOWING TWENTY-FIVE CYLINDRICAL BOILERS.

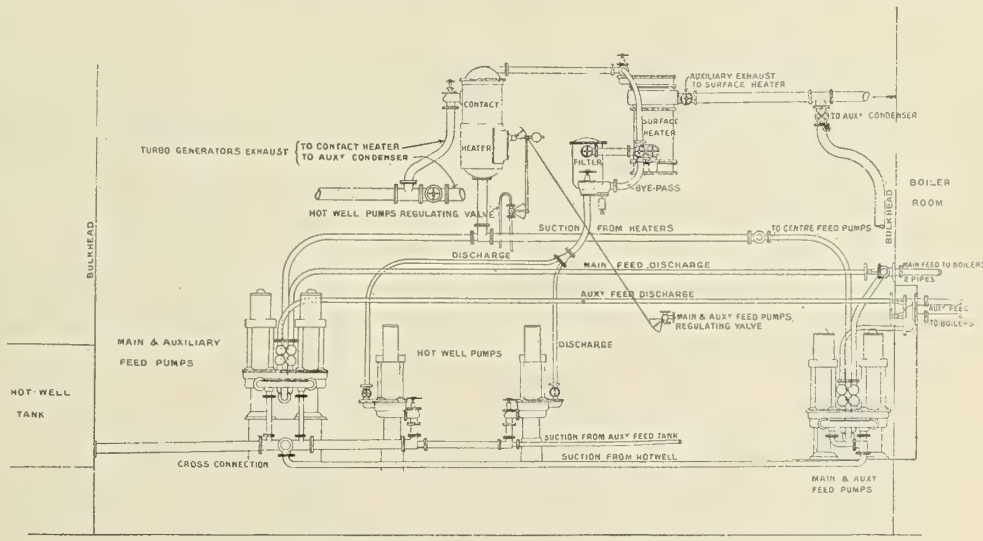


FIG. 4.—DIAGRAM OF FEED HEATING AND SUPPLY SYSTEM, STEAMSHIP LUSITANIA.

atmosphere. The turbo-generators exhaust into an entirely distinct system of piping connected to the contact feed heaters, auxiliary condensers and atmosphere. From Fig. 4 it will be seen that the hot-well pumps draw from the hot-well and discharge through two feed filters to the two surface feed heaters, and thence to the contact heaters; these latter constitute the usual feed receivers, and are fitted with float gear, controlling not only the steam supply to the main feed pumps, but also to the hot-well pumps. In addition, the hot-well pumps are controlled by float gear in the hot-well tanks—the latter, in case of a shortage of water supply, to prevent the pumps running away, and the former precluding the possibility of water being forced back to the turbo-generators, should the non-return valve in the steam inlet to the contact heater fail to act in the event of the feed pumps suddenly slowing.

In most large merchant ships, on account of accidents which have occurred, it has been the general practice to lead the exhaust from the electric engines direct to the main or auxiliary condensers only, to insure steady running of the dynamos, and to prevent the possible passing back of water from other auxiliary engines into the cylinders of the electric engines. In this present case the electric installation is so large, and the temperature of the hot-well, due to the high vacuum maintained, so low, that it was considered to be essential to utilize this exhaust for the heating of the feed water, and, by adopting this duplicate system of feed heaters and hot-well control, this end has been attained with a minimum of risk.

Owing to the very great length of feed supply pipes from the feed pumps to the forward boilers, there was considerable speculation as to whether the feed supply to each boiler room would have to be entirely distinct. Had this been so, each pair of feed pumps would have required to be worked entirely independently of the others, and, in the event of any hitch occurring, one of the two pairs of stand-by pumps would have had to be at once connected to the particular boiler room in question. In actual running, however, it was found that the four main feed supply pipes could be cross-connected and made common, thus making any demand for special care and vigilance on this score unnecessary.

With regard to the maintenance of steady steam at sea, although innumerable types of rocking and self-cleaning fire-bars have been devised and tried in connection with automatic stokers, none of these has been found to give satisfactory results in merchant vessels. The stoking, and especially the cleaning of the fires whenever hard steaming is required, is the same as it was fifty years ago, and, as a consequence, is dependent on the willingness and ability for work of each unit of the stoking complement of the ship.

(To be Concluded.)

THE SECOND CLASS CRUISER GLADIATOR, SUNK IN COLLISION WITH THE ST. PAUL.

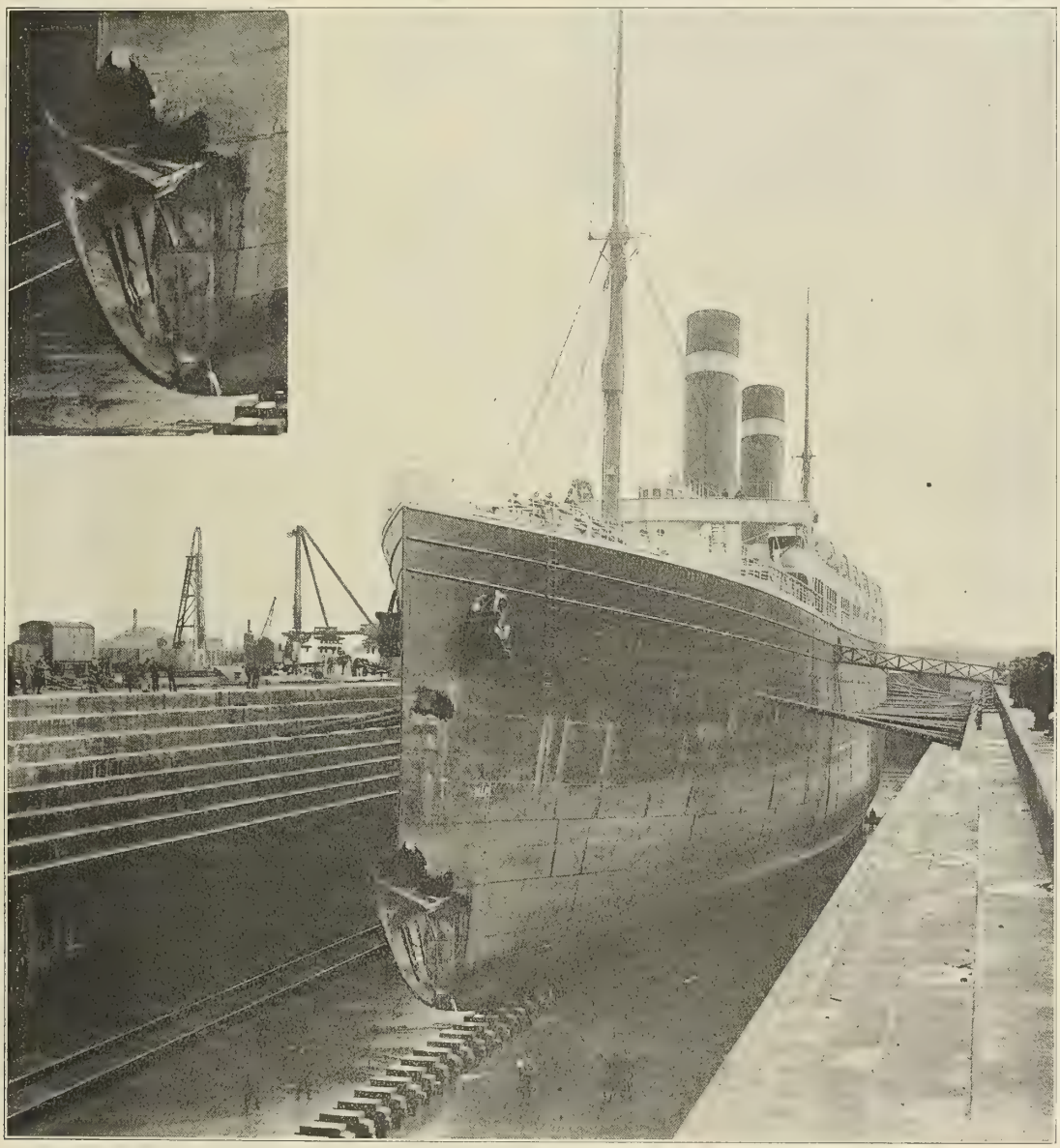
THE SINKING OF THE GLADIATOR.

On April 25 occurred a disaster almost without parallel in naval history, in that a warship in collision with a merchantman was sunk. The instances of this sort have been extremely rare, the greater damage coming usually upon the more lightly plated vessel of commerce. In the present case, however, the American Line steamer *St. Paul,* which had just left Southampton for New York, and was proceeding at slow speed through the Solent in the midst of a heavy snowstorm and fog, rammed the British cruiser *Gladiator* on the starboard side between the bridge and the forward funnel, to such effect that within about twenty minutes the *Gladiator* touched bottom. Fortunately, the swing of the tide, aided by her engines, sent the vessel close to shore before she finally went down, and a considerable portion of the hull, including one of the propellers, remained above water, as she rested on her beam ends. One of our photographs shows her in this condition.

The liner cut clear through the side of the war vessel, and

THE WRECK OF THE GLADIATOR—THE SHIP ON HER BEAM ENDS—BATTLESHIP PRINCE GEORGE IN BACKGROUND.
(Photograph, Stephen Cribb, Southsea.)

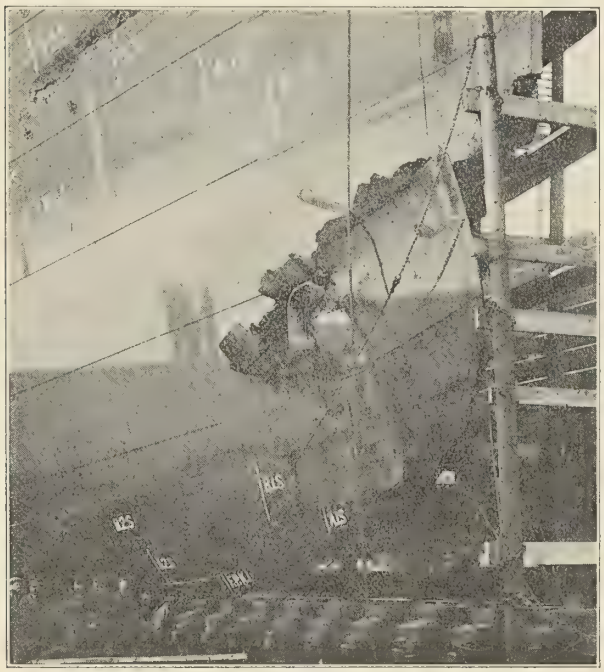
in backing away carried off one plate of the protective deck (weighing considerably more than a ton), a watertight door, some coal bunker casings, and a number of sections of angles joining these plates together. These were left embedded in the hole made in the liner's bow by the resistance of the protective deck of the cruiser, and are clearly shown in the small detail in the upper corner of one of our illustrations. The damage to the liner was considerable, but she was in no



THE AMERICAN LINER ST. PAUL IN DRY DOCK AT SOUTHAMPTON, SHOWING DAMAGE TO BOW.

danger, because of the fact that her forward collision bulkhead intervened between the apertures and the main part of the hull. On the starboard side, however, indentations, including even the splitting of plates, ran back some 45 feet from the bow above the waterline. These were due, doubtless, to projecting guns, or something of that sort, on the cruiser.

The impact was almost at right angles. At the request of the captain of the cruiser, the *St. Paul* was backed out of the



CLOSE VIEW OF THE DAMAGED BOW OF THE ST. PAUL.

gap instead of being allowed to block the gap and push the cruiser ashore into shallow water. As soon as the vessels were clear, the *Gladiator* began to list heavily to starboard, and took the ground in about 6 fathoms, at a quarter of a mile from shore. The *St. Paul* lowered boats to assist in the saving of the crew of the *Gladiator,* many of whom were injured, and four or five of whom actually boarded the liner at the moment of collision. Unfortunately, these boats were swept far away by the tide and wind before they could be used, and those of the *Gladiator,* being launched from her high side, fell against the side of the vessel, and were with difficulty put afloat. The loss of life is said to have been twenty-eight, including one officer, all of whom belonged to the *Gladiator.*

The cruiser was launched in 1896 at Portsmouth, having a displacement of 5,750 tons, a length between perpendiculars of 320 feet, a beam of 57 feet 6 inches and a draft of 24 feet. With twin-screw engines of 10,000 horsepower the vessel made 19 knots. Steam was furnished by Belleville boilers. The battery included ten 6-inch guns and eight 12-pounders, besides six smaller guns and three torpedo tubes.

The *St. Paul* is a twin-screw passenger steamer, launched in 1895 by William Cramp & Sons, Philadelphia. She measures 535 feet 6 inches between perpendiculars, with a beam of 63 feet and a draft of 26 feet 8 inches. Her registered tonnage is 11,629. She has quadruple expansion engines of 20 000 horsepower, giving a maximum speed of 21.5 knots.

The battleship *Michigan* was launched by the New York Shipbuilding Company, Camden, N. J., May 26. She is the first American "all big gun" battleship, has a displacement of 16,000 tons, and the main battery will consist of eight 12-inch rifles, all training on one broadside.

RESULTS OF FURTHER MODEL SCREW PROPELLER EXPERIMENTS.*

BY R. E. FROUDE, F. R. S.

GENERAL SCOPE OF EXPERIMENTS.

The experiments on model screw propellers, with which this paper is concerned, are an extension of those made at the Torquay tank in 1884. Since the present experiments, while covering a good deal of fresh ground, covered also the whole ground of the previous ones—so far, at least, as is relevant to present-day conditions—and were carried out, in some ways, on improved methods, their results may be said to supersede those of the earlier experiments.

The principal points in which the present experiments exceeded in scope the previous ones were as follows:

(i.) Lower values of pitch ratio included. [The highest values included in the old experiments were at the same time discarded; but these were quite outside the range of present practice.] This downward extension of the pitch-ratio range in the new experiments brings in pitch ratio as an important factor affecting efficiency, since it is in the region below the range of the old experiments that pitch ratio influences efficiency in an important way.

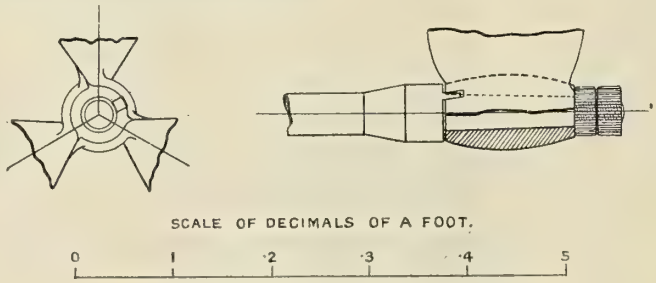


FIG. 4.—BOSS OF MODEL PROPELLERS.

(ii.) Three-bladed as well as four-bladed propellers were tried [in the old experiments the model propellers were all four-bladed].

(iii.) Variation of width proportion of blade was tried, ranging from that of the old experiments up to about twice that width.

(iv.) Difference in blade shape; by including the wide-tip pattern, in addition to the elliptical pattern solely used in the old experiments.

For outlines and sections of blades and shape of boss, etc., see Figs. 1 to 4. For elliptical propellers, the developed outlines are ellipses of major axis equal to propeller radius, and the blade-width ratio is the ratio of minor to major axis. The developed outlines of the wide-tip propellers are formed from elliptical outlines for the same reputed blade-width ratio by making, for any radius, $= r$,

$$\frac{\text{Wide-tip width ordinate}}{\text{Elliptical width ordinate}} = \left[\frac{1}{2} + \frac{r}{R}\right]$$

where $R =$ propeller radius. Hence, for the same blade-width ratio, the wide-tip and elliptical outlines have the same total area. For the purpose of the analysis, and for relating disk-area ratio to blade-width ratio, the entire area of the elliptical or wide-tip outline thus obtained is reckoned, without any allowance for portion covered by boss.

METHOD AND GENERAL CONDITIONS OF EXPERIMENT.

Method.—The experiments were the usual kind of what are described as screw experiments "in open," namely, in undisturbed water (without model in front), the screw being mounted on the forward end of the shaft which drives it, the whole advancing through the water at a prescribed speed, the

* Read before the Institution of Naval Architects, London, April 10, 1908.

PARTICULARS OF REGULAR SERIES OF MODEL PROPELLERS.

Reference No.	Blade Width Ratio.	Disk Area Ratio.	Pitch Ratio.
	THREE-BLADED ELLIPTICAL.		
1	.4	.3	.885
2	.55	.413	.885
3	.7	.525	.885
4	.4	.3	1.09
5	.55	.413	1.09
6	.7	.525	1.09
7	.4	.3	1.315
8	.55	.413	1.315
9	.7	.525	1.315
10	.4	.3	1.53
11	.55	.413	1.53
12	.7	.525	1.53
	THREE-BLADED, WIDE TIP.		
13	.6	.45	.885
14	.8	.6	.885
15	1.0	.75	.885
16	.6	.45	1.09
17	.8	.6	1.09
18	1.0	.75	1.09
19	.6	.45	1.315
20	.8	.6	1.315
21	1.0	.75	1.315
22	.6	.45	1.53
23	.8	.6	1.53
24	1.0	.75	1.53
	FOUR-BLADED, ELLIPTICAL.		
25	.4	.4	.885
26	.55	.55	.885
27	.7	.7	.885
28	.4	.4	1.09
29	.55	.55	1.09
30	.7	.7	1.09
31	.4	.4	1.315
32	.55	.55	1.315
33	.7	.7	1.315
34	.4	.4	1.53
35	.55	.55	1.53
36	.7	.7	1.53

screw driven at prescribed revolutions per minute, and the thrust and turning moment being measured. Each set of experiments on each screw consisted of some twenty runs at the same speed, at different slip ratios; including a run or two, at beginning and end, with screw off, for eliminating constant resistances of apparatus. This series of runs, which occupied some 2½ hours in the middle of each day, was daily both preceded and followed by a series of experiments, with truck stationary and a turbine brake dynamometer substituted for the screw, to measure the "working" friction of the driving gear.

Diameter of Screws.—Uniformly 0.8 foot [as compared with 0.68 foot in the old experiments].

Immersion (to center of shaft).—Uniformly 0.64 foot [viz., 0.8 of the diameter, as in the old experiments].

Speed of Advance.—300 feet per minute [as compared with 206 feet per minute in the old experiments]. This was the highest speed at which it would have been possible to obtain the desired range of slip ratio, without overstraining the driving gear.

Boss of Screws.—The boss common to all the model screws (see Fig. 4) was (as usual in our model screws) of the smallest practicable character. Some special experiments made with ordinary large bosses showed the effect of a large boss to be material, partly in its resistance, and still more in an increase of revolutions per minute, due presumably to stream line action; but, as both these effects are intimately involved with those of the bearings or shaft tubes, which in the ship immediately precede the propeller, it seems best, where all these features are absent, as in the experiments with which we are here concerned, to suppress the boss so far as possible.

Skew-Back of Blades.—The model screws employed in the regular series of experiments were without skew-back; but experiments were made with four additional model screws having a skew-back of 15 degrees on the driving face, otherwise similar to four of the regular series, and the skew-back was found to make no material difference in the result. These four skew-back screws were generally similar to Nos. 7, 8, 9 and 31 of the list.

ANALYSIS AND REDUCTION.

Basis.—It is perhaps in the system of analysis and reduction of the results, more than anything else, that the present series

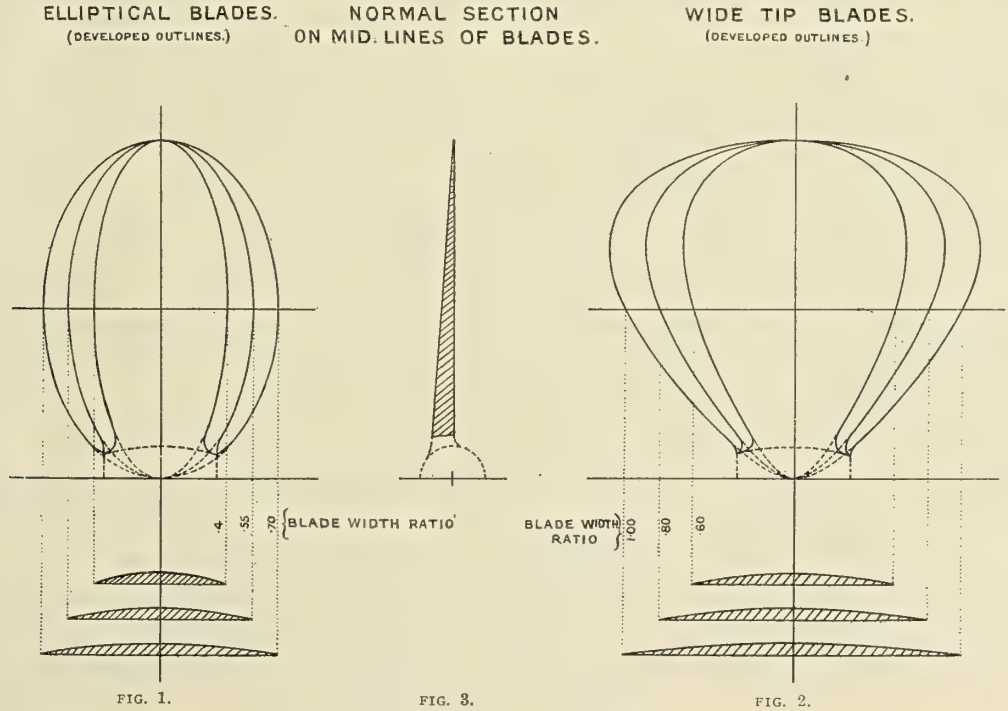


FIG. 1. FIG. 3. FIG. 2.

of experiments differs from the previous. The introduction of width proportion and shape of blades as varying conditions, and the recognition of pitch ratio as having an important influence on efficiency, in any case necessitated some change; but also, the possibility had presented itself to me, in course of study of the results of the previous experiments, of so harmonizing the results that the relation of thrust to revolutions per minute could be calculated for a screw of any diameter and pitch, by means of a simple formula, without using any diagrams or tables at all.

Apart from the convenience of being able to embody the results in such a form, it is, of course, always desirable, when possible, in harmonizing the minor and accidental irregularities of a system of experiment results, to use a mathematical expression which has some at least approximate theoretical relevance, in place of a mere graphic fairing of spots by a curve. The analysis, then, of the series of thrust and efficiency curves yielded by the experiments on the individual model screws, was in the first instance based on the following simple formula for thrust in terms of revolutions per minute:

$$T = a\ R^2 - b\ R \qquad (1)$$

Where T = thrust; R = revolutions per minute; a = a coefficient depending on dimensions, etc., of screw; b, one depending on the speed and pitch.

This formula embodies the following idea, which under certain ideal conditions would be theoretically correct:

Imagine an elementary blade mounted on the end of a resistless spoke, revolving at circumferential speed R, and standing at an angle α from the plane of rotation; the whole having

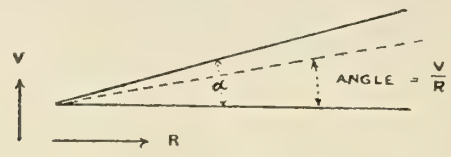


FIG. A.

an axial speed V relatively to the fluid. Then (see Fig. A), assuming α "small," if R_0 be the circumferential speed for zero thrust, we may put

$$a = \frac{V}{R_0},$$

and

$$\text{Thrust} = T = (\text{say})\ k\ R^2\left(a - \frac{V}{R}\right) = \left(k - \frac{V}{R_0}\right)R^2 - (kV)R.$$

In any screw, revolving in still water at various rotary speeds without axial advance, the thrust will, of course, be proportional to the square of the rotary speed. This fact is expressed by the term $a\ R^2$ of the formula, represented by the ordinates of the parabola $A\ B\ C\ D$ in Fig. B. If now we suppose the screw, while still revolving as before at various rotary speeds, to have a definite forward axial linear speed of advance V as well, there will, of course, then be a certain definite rotary speed (revolutions per minute = R_0, say), at which the thrust will be zero, below which it will be negative, and above which it will be positive, but of decreased amount. And the formula expresses this decrease of thrust in terms of revolutions per minute, by the negative term $b\ R$, represented by the ordinates to the straight line $A\ C\ E$, cutting the parabola at C, namely, at R_0, the revolutions of zero thrust. Thus the straight line $A\ C\ E$ becomes in effect the zero line for the curve $C\ D$, regarded as the thrust curve for the speed of advance V; and, similarly, the same parabola $A\ B\ C\ D$ may be made to furnish the thrust curve for the same screw at any speed, by drawing accordingly the sloping line $A\ C\ E$ which represents the term $(-b\ R)$ of the formula.

Even before the experiments herein described were begun, I had found that this formula expressed the thrust curves of experiment, in general, remarkably well; since, therefore, a new reduction was in any case necessary, this formula was unquestionably the right basis for, at any rate, the primary analysis of the results.

Thrust.—Continuing the study of the formula, it will be seen that if, as is most convenient for such analysis, and as has been done in this case, we take as a conventional measure of the pitch the travel per revolution at the revolutions per minute R_0 (of zero thrust), so that

$$P = \frac{V}{R_0},$$

and, again, observing that for zero thrust at revolutions $= R_0$ we must have $a\ R_0^2 = b\ R_0$, we can eliminate the coefficient b in terms of a, V and P. And, assuming the coefficient a to have

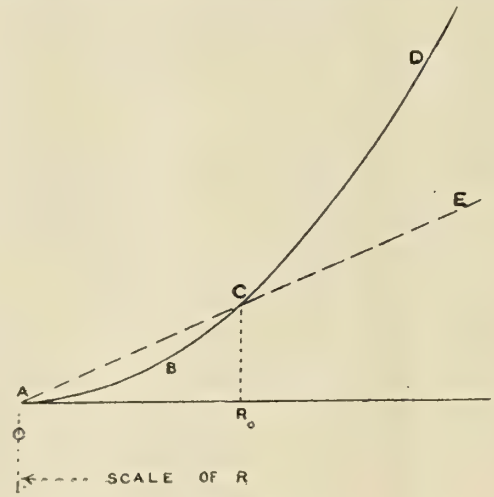


FIG. B.

been correctly obtained for a screw of specific design and unit diameter (diameter = D = 1), we obtain from equation (1) above, the following two alternative equations for thrust:†

$$T = \frac{a}{p^2} D^2\ V^2 \frac{S}{(1-S)^2} \qquad (2)$$

$$T = a\ D^4\ R^2\ S, \qquad (3)$$

where

$$p = \frac{P}{D},$$

or pitch ratio, and S = slip ratio as ordinarily reckoned, viz.: $(R - R_0) \div R$. The former of these, expressing thrust in terms of speed and slip ratio, is perhaps the most intelligible; while the latter is often more convenient for computation.

To enable the thrust, for given speed and revolutions, to be

† Substituting, in equation (1), $a\ R_0$ for b, we get—

$$T = a\ R^2 - a\ R_0\ R = a\ R^2\left(1 - \frac{R_0}{R}\right) = a\ R^2\ S.$$

This is for unit diameter. For propellers of uniform design, area increases as (diameter)^2; and, for given revolutions, so does thrust per unit of area as affected by speed; thus giving equation (3) of text. Next, remembering that—

$$(1 - S) = \frac{V}{R\ P},$$

and substituting for R in equation (3), we get—

$$T = a\ D^2\ V^2 \frac{D^2\ S}{P^2\ (1-S)^2} = \frac{a}{p^2} D^2\ V^2 \frac{S}{(1-S^2)}.$$

N.B.—In the analysis of the model propeller results, V has been taken in hundreds of feet per minute, and R in hundreds of revolutions per minute.

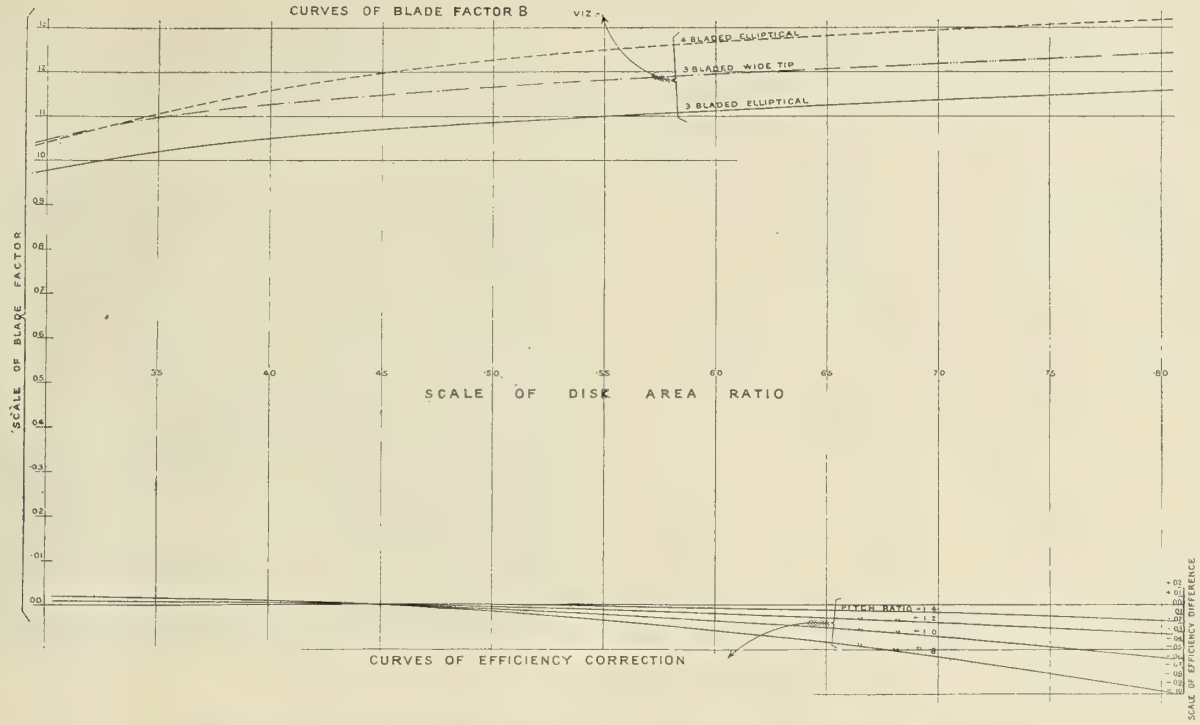


FIG. 5.—CURVES OF BLADE FACTOR AND EFFICIENCY CORRECTION ON BASE OF DISK AREA RATIO.

calculated by either of these formulæ for a propeller of any given design and dimensions, it is needed only to determine the coefficient a as affected by difference of design, the principal elements in which may be taken to consist in pitch ratio, type and blade width proportion.

It was found that the effects of these two principal elements might be taken as independent of one another, and that, as regards pitch ratio, a might be most correctly taken as proportional to $p\ (p + 21)$. As regards type, the value

$$\frac{a}{p\ (p + 21)}$$

, which, as just seen, is constant for varying pitch ratio, was taken as the expression for the "blade factor" B; the purpose of which is to denote what may be called the thrust capacity of the propeller, as dependent on type, *i. e.*, whether three-blade elliptical, three-blade wide-tip, or four-blade elliptical; and within each of these types, on width proportion of blade. The value of this blade factor B, as obtained from

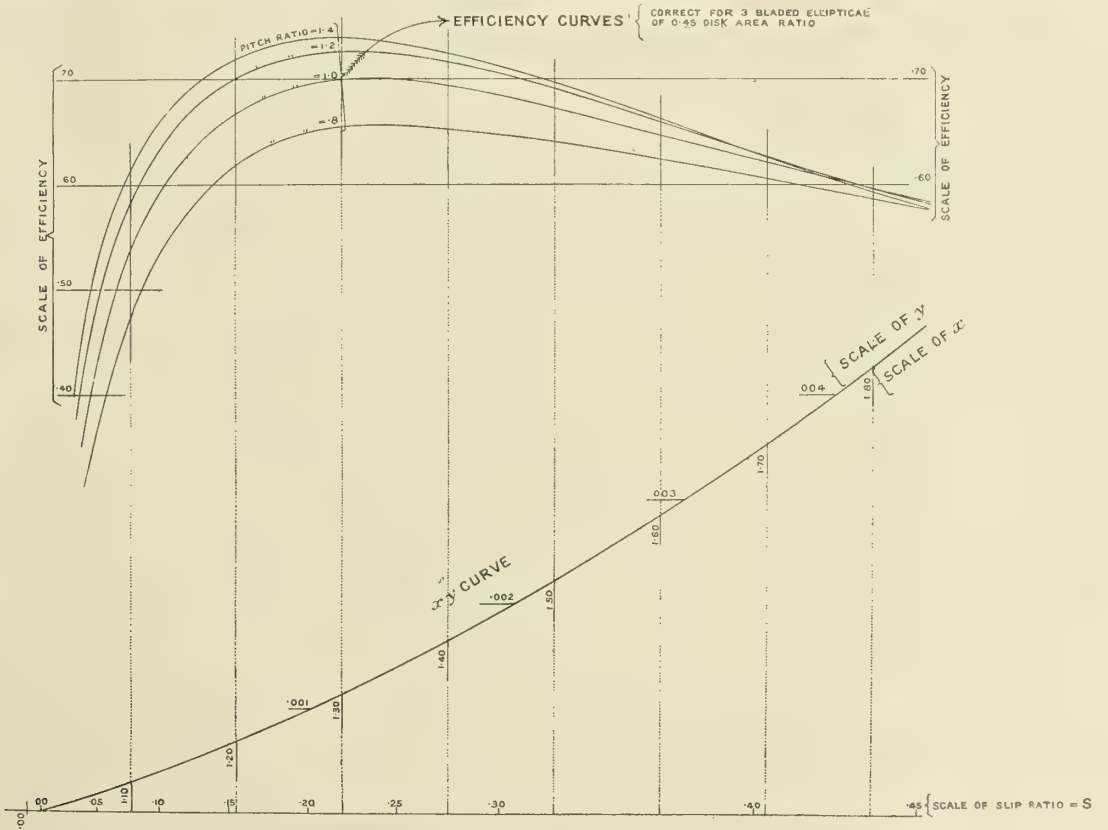


FIG. 6.—EFFICIENCY CURVES AND X-Y CURVE BASED ON SLIP RATIO.

the experiments, and as dependent on these variants, has been indicated by the ordinates of three curves (see Fig. 5) respectively proper to the three types just mentioned, on an abscissæ scale representing blade width proportion, as indicated by "disk area ratio," namely, ratio of total blade area to disk area.

At the same time, for a final test of the formula of equation (1), as accurately expressing the variation of thrust with revolutions, the thrust values of all the individual propellers, at a series of slip ratio values, were carefully compared with thrust values calculated by formula; and on this information the thrust formula was corrected by multiplying the right-hand side by 1.02 (1 — .08 S). Making this correction, and also substituting for a its value in terms of the blade factor B, just referred to, equation (2) becomes the final thrust formula, as follows:

$$T = D^2 V^2 \times B \frac{p + 21}{p} \times \frac{1.02\ S\ (1 - .08\ S)}{(1 - S)^2}. \qquad (4)$$

To facilitate calculations, a curve was computed expressing the last factor (involving S only) as an ordinate ($= y$) to a base ($= x$), expressing revolutions and pitch relatively to speed, as indicative of the slip ratio S. This curve, commonly called the "x y" curve, appears in Fig. 6. Conveniently for ship screw calculations, the numerical coefficients used in the computation of this curve were chosen for expressing, not thrust, but "thrust horsepower" (or $T\ H\ P$) $= H$; speed $= V$, in knots; revolutions $= R$, in hundreds; diameter $= D$, in feet. We thus get as the expressions for x and y as follows:

$$x = \frac{R\ p\ D}{V} \left[= \frac{1.0133}{1 - S} \right] \qquad (5)$$

$$y = \frac{p}{B\ (p + 21)} \times \frac{H}{D^2\ V^3} \left[= .0032162 \frac{S\ (1 - .08\ S)}{(1 - S)^2} \right] \qquad (6)$$

In regard to the expression for x

$$1.0133 = \frac{6.08}{6} = \frac{\text{speed in hundreds of feet per minute}}{\text{speed in knots.}}$$

In regard to that for y; from equation (4),

$$\frac{T}{D^2\ V^2} \left[= \frac{T\ V}{D^2\ V^3} \right] = B \frac{p + 21}{p} \cdot \frac{1.02\ S\ (1 - .08\ S)}{(1 - S)^2}.$$

Then, converting V from 100's feet per minute into knots, and $T\ V$ into H, this equation becomes

$$\frac{H \dfrac{33{,}000}{100}}{D^2 \left(\dfrac{101.33}{100} \right)^3} \times \frac{p}{B\ (p + 21)} = \frac{1.02\ S\ (1 - .08\ S)}{(1 - S)^2}.$$

or

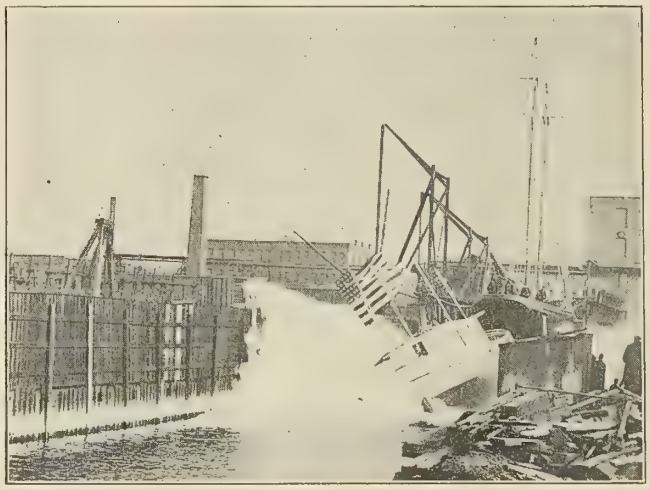
$$\frac{H}{D^2\ V^3} \times \frac{p}{B\ (p + 21)} = .0032162 \frac{S\ (1 - .08\ S)}{(1 - S)^2}.$$

(To be Concluded.)

An unusually large number of new ships have visited New York during June. They include the Holland-America liner *Rotterdam,* 24,170 tons; the French liner *Chicago,* 11,000 tons; the Russian-American liner *Russia,* 14,000 tons; the North German Lloyd liner *Prinz Friedrich Wilhelm,* 17,500 tons, and the *Principe de Udine,* of the Lloyd Sabaudo.

The Tug Monomack.

This tug was launched from the yard of the Atlantic Works, East Boston, on March 7. She took her maiden plunge sideways into a slip which is just wide enough to permit the travel of a marine railway cradle. The boat is owned by the Merri-



THE TUG MONOMACK GOING OVERBOARD.

mack River Towing Company, and is to be used for general towing. Her length over all is 77 feet 6 inches, with a beam of 17 feet 2 inches; depth of hold, 7 feet 8 inches, and a draft of 7 feet 6 inches. H. M. Smith.

Transatlantic Steamship Records.

The first steamship record of any vessel crossing the Atlantic was the twenty-five days occupied by the *Savannah* in 1819. Nineteen years later the *Great Western* crossed from Liverpool in fourteen days. In 1840 the Cunard steamer *Britannia* reduced the record to ten days and six hours. In 1852 the Collins Liner *Arctic* made the trip in nine days and seventeen hours. Other records are as tabulated:

	Days.	Hrs.	Min.
1856—*Persia* (Cunard)	9	1	45
1866—*Scotia** (Cunard)	8	2	48
1869—*City of Brussels* (Inman)	7	22	3
1875—*City of Berlin* (Inman)	7	15	48
1877—*Britannic* (White Star)	7	10	54
1880—*Arizona* (Guion)	7	7	23
1882—*Alaska* (Guion)	6	18	37
1884—*Oregon* (Cunard)	6	11	9
1885—*Etruria* (Cunard)	6	4	43
1889—*City of Paris* (Inman)	5	19	18
1891—*Majestic* (White Star)	5	18	10
1891—*Teutonic* (White Star)	5	16	31
1893—*Paris* (American)	5	14	24
1893—*Campania* (Cunard)	5	12	7
1907—*Lusitania* (Cunard)	5	..	54
1907—*Lusitania* (Cunard)	4	18	40
1908—*Mauretania* (Cunard)	4	20	15

This does not include records from Southampton or Plymouth.

* The last of the paddle-wheel steamers.

The Largest Ships in the World.

A German contemporary has listed all the vessels of over 20,000 tons now afloat or building, there being nine in service, one other (the *Rotterdam*) very nearly ready, and four

others in various stages of construction or "anticipation." This makes a total of fourteen. Four of the nine in service, and one of those under construction, are accredited to the White Star Line, all having been built by Harland & Wolff, of Belfast. The Hamburg-American Line is accredited with two in service and two under construction, three of which total were built by Harland & Wolff; the other was built by the Vulcan Works at Stettin. The Great Northern Steamship Company has one vessel, built in the United States; the Cunard Line has two ships, built on the Clyde and the Tyne, respectively; the Holland-America Line has one ship, practically finished, by Harland & Wolff, and the North German Lloyd Line is having one large vessel built by the Vulcan Works.

Omitting the four vessels on which only slight progress has been made, we find ten immense steamers, aggregating 246,309 tons gross, or an average of 24,631 tons. Six fly the British flag, two are German, one American and one Dutch.

that the contour of drawing and that of the pattern shall be exactly similar. If such is not the case, why exercise such care in reading the areas, the ratios, etc.? It seems to be the rule for the drawing to show one shape and the pattern another. Hence, as the developed area shown on the drawing and the area of blade are different from those in metal, of what use are all the refinements into which designers enter in designing a wheel? If we take a drawing of a wheel, and, after the ship has undergone her trial trip, compute from the drawing the characteristics of the blade, and use them in the computation for data required in such cases; then, if the wheel as built is not of the shape as shown on the drawing, of what use would there be in using this information for designing other ships? The writer had this very forcibly impressed upon him some time ago.

Fig. 1 shows a wheel where the drawing and pattern were exactly similar. Fig. 2 shows the same blade, as drawn by a well-known method. It is not the object of this paper to dis-

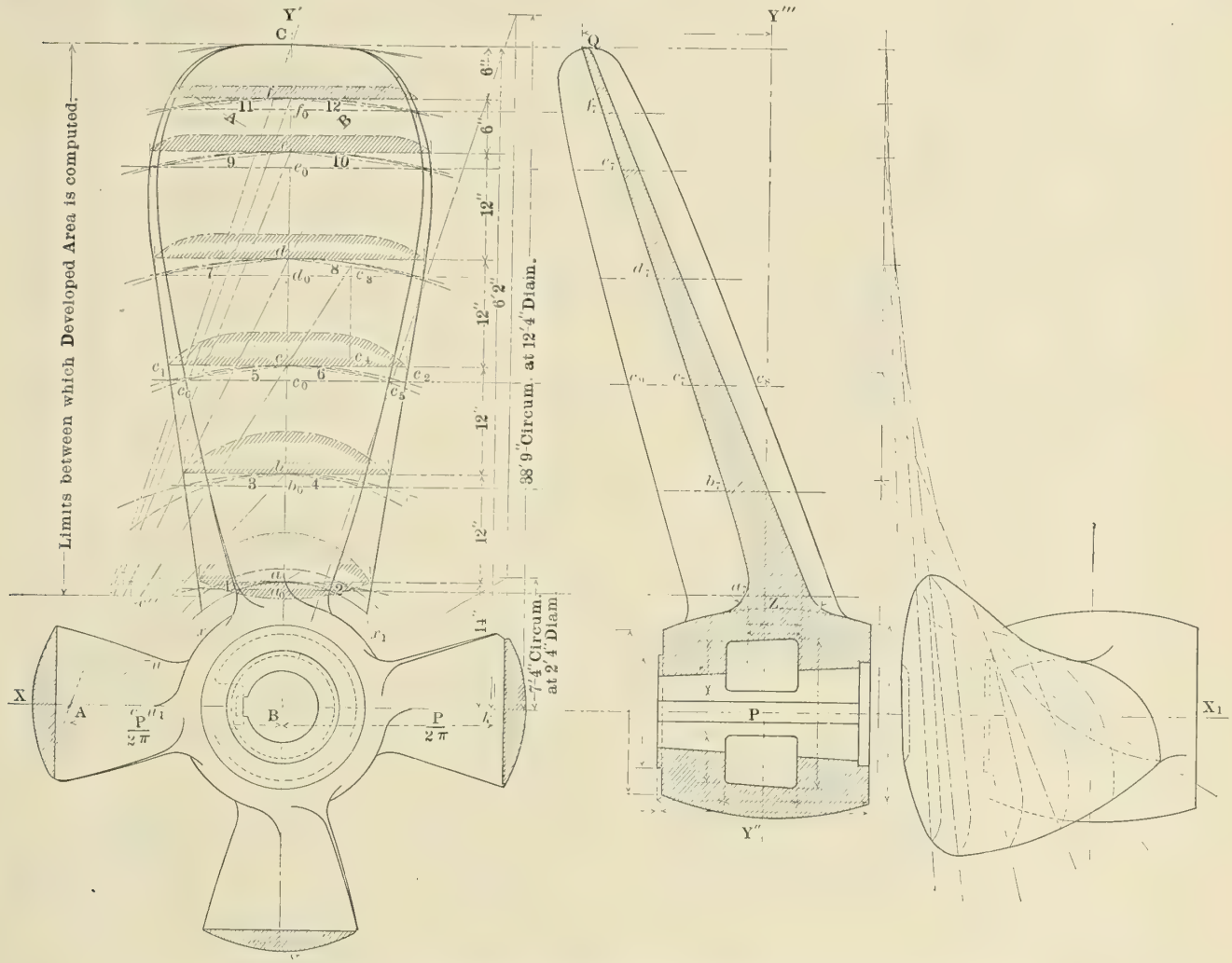


FIG. 1.—THE DEVELOPMENT OF A PROPELLER WHEEL OF 12 FEET 4 INCHES DIAMETER AND AN EQUAL PITCH.

THE LAYING OUT OF PROPELLER WHEELS.

BY CHARLES S. LINCH.

There is no one subject within the limits of the science of naval architecture which offers more interest to the investigator than the propeller wheel. The mist which has obscured the subjects of resistance and propulsion is rapidly vanishing before the investigations carried on at the experimental tanks, under such investigators as Durand, Taylor, Rota and Froude.

It is not only with regard to these two subjects that diversity of opinion exists, but in the delineation of the wheel as well. It should be the aim of the designer to so delineate the wheel

cuss the various methods or to criticise their faults and virtues, but to submit to the readers of this journal a method of delineating the blade, and a method of making the pattern whereby the two agree in every detail.

Let us take the wheel shown in Fig. 1. This wheel is 12 feet 4 inches in diameter and 12 feet 4 inches in pitch. We will assume that we are given the required developed area, and that we have gone through the preliminary work, and have decided on shape of blade as shown, and are prepared to make the working drawings for same.

We first draw the horizontal line XX_1. This line is the axis

of our shaft. Now draw a line $Y\,Y'$ at right angles to $X\,X_1$, intersecting the axis in B. About this center line will be drawn our athwartship view, or in other words, these lines lie in an athwartship plane. From the center B we set off on either side of the vertical, or its intersection, a distance equal to

$$\frac{\text{Pitch}}{2\pi} = \frac{P}{6.2832}$$

From B we set off on $Y\,Y'$ a distance $B\,C$, equal to the radius of the wheel. In our case this is 6 feet 2 inches. Now,

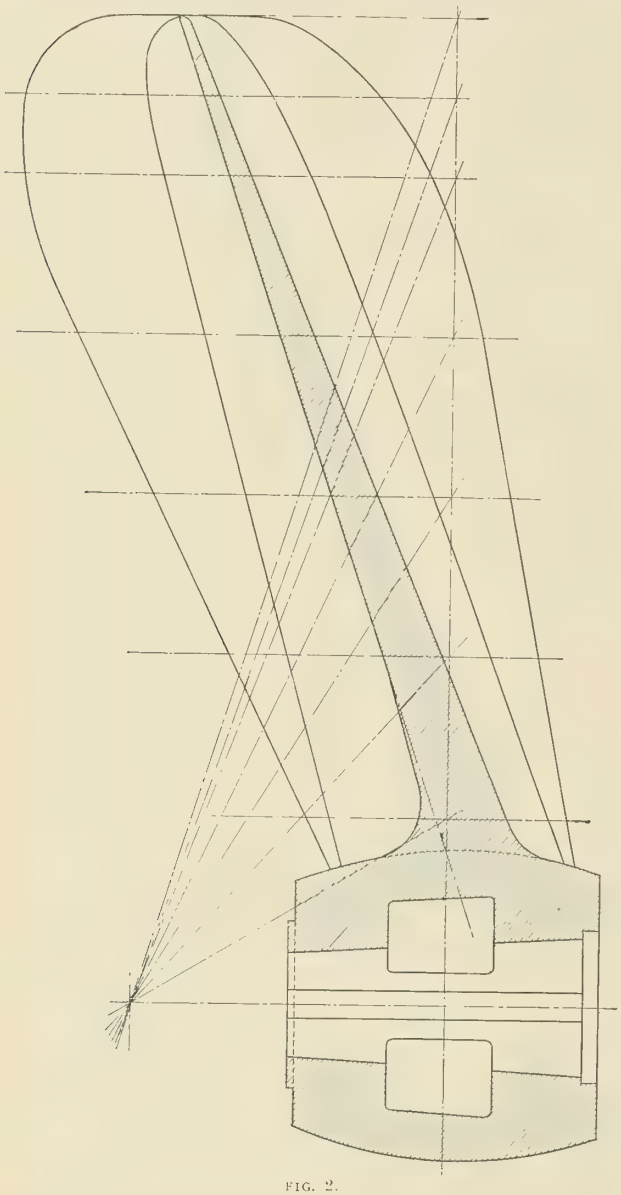


FIG. 2.

from B we set off a distance equal to the radius of the hub, or we can ignore the hub for the present, and set off on $Y\,Y'$ from the point B the distance Ba, Bb, Bc, etc., as shown. It is better to decide on the hub and to sketch in its contour.

From the points a, b, c, etc., we draw a series of lines to A (these lines are termed pitch lines), and through the points a, b, c, etc., we describe elliptic arcs, the radii of these arcs all lying on $Y\,Y'$ produced. Set off from a, b, c, etc., the *required distances*, as 1, 2, 3, 4, etc., these distances being obtained from our preliminary work, where account is taken of the requirements for blade area. Now pass a fair curve through these points, and we obtain the contour of the developed blade. The area included within this boundary should be the developed area of the wheel as built, except in so far as slight variations would modify it.

With B as center and radius Ba, Bb, Bc, etc., describe circular arcs (the elliptic arcs project on an athwartship plane into circular arcs). We can now proceed to lay down the projected area by one of two methods. We can, from the points c_1 c_2, project a horizontal line parallel with the axis, until it intersects the circular arc, and thus get points for the projected area; or we can simply produce the pitch line, for instance, through c, and take a distance c_3 c_4, equal to c' c'', and lay off from c_0 the distances c_0 c_5, c_0 c_6, proceeding thus for each arc, the result in either case being precisely the same. Take the former method. Project back from c_2 to c_5, and repeat for each arc, passing a curve through the points thus obtained. We have at once the contour of the projected view, and the area inclosed will give the projected area. Now for the fore-and-aft plane.

Draw the perpendicular $Y''\,Y'''$ intersecting the axis at P. Set off $Y'''\,Q$ equal to the "fall back" or skew. From Q draw a straight line QZ, intersecting $Y''\,Y'''$ in Z. [Note—It is simply the desire of the man having the wheel made that the generating line shall intersect at this point. It is not any easier for the pattern maker, and there is no reason why it should not intersect $Y''\,Y'''$ at P.]

From the points of intersection of the elliptic arcs with the boundary line of the blade on the athwartship plane, draw horizontal lines parallel with the axis to a_7 b_7 c_7, etc. Set off on either side of the generating line $Q\,Z$ the distance $A\,u$ or $c\,c_4$. This gives the point of division, and passing a curve through these points we have the contour of blade when viewed from a vertical plane parallel with the shaft.

The thickness of the blade at tip and root having been decided, as well as the successive blade sections in the preliminary work, we proceed to draw them in, and the remaining details can now be filled in. We have used the elliptic arc for our developed area. It will be shown later that the pattern is built up in precisely the same way.

Now to prove the correctness of the construction and the reasoning. It is evident that a straight line moving uniformly around an axis which it intersects at right angles (or inclined at any angle), and advancing uniformly along the axis will sweep out a surface. This surface is termed a helicoid. The first line is termed the generating line or generatrix; the second the axis. Now it is apparent that any point in the generating line will trace out a helix. In other words, the locus of any point in the generatrix, as it moves in obedience to the law expressed above, will be a helix. Keeping in mind the idea of the surface, we see that the locus of the point must be in the surface.

If, for one complete revolution of the generatrix, every point moves through equal distances in the direction of the axis, the pitch will be uniform, and hence the surface is a surface of uniform pitch. This is the surface used for deriving the face of a blade of uniform pitch. It can be proved that a helix traced by any point in the generating line is a curve of intersection of a cylinder with the screw surface. Take any number of points on the generatrix; each of these points will limit the radius of a cylinder. The cylinders, having the same axis, are termed co-axial.

We observe from Fig. 3 that as the diameter of the cylinder increases, the pitch being constant, the angles of the helix decrease.

Let P = pitch; r = radius of cylinder; α = angle of helix.

Then $\tan \alpha = \dfrac{P}{2\pi r}$

Now, co-axial cylinders intersect the screw surface in helices, and, as we have seen, make certain angles with the

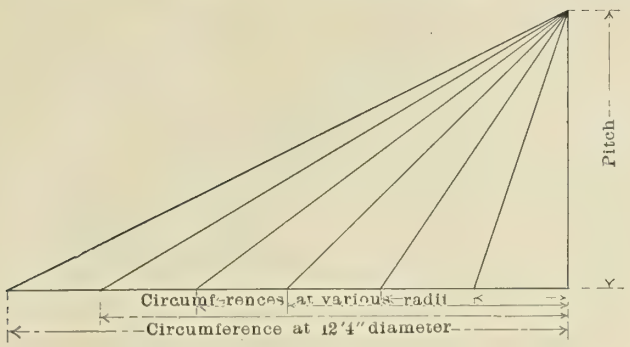


FIG. 3.

axis. If we pass planes through the middle point of the helical curve, keeping the same angle with the axis, we find that they will intersect these planes in elliptic arcs. In the delineation of the blade, we swing these elliptic arcs at all radii around the common center line *B C* until they all lie in the same plane. This, of course, makes their major and minor axes coincident; but, as shown, they are of different length. We can either compute the radii of these elliptic arcs, or we can obtain them by construction.

Take the arc passing through *b*, for example. We have seen that the distance *Bb* is the semi-minor axis. The distance *bA* is the semi-major axis. The distance $\frac{P}{2\pi}$ locates the foci of all the elliptic arcs. Let $oy = Bb$, and $ox = bA$, Fig. 4. Complete the triangle. Produce *oy* in both directions. With *ox* as distance, set off on *oy* produced, $oy' = o\ x$. Set off on *yx*, from *y*, the distance $y\ y' = yu$. Now, $y\ u = y\ y' = o\ x - o\ y =$ difference between semi-major and semi-minor axis. Bisect *u x*, and produce the perpendicular to intersect *o y* produced, as at *w*; then *w y* is the radius of the elliptic arc.

To prove the correctness of this construction. In the elliptic arc passing through *b*, Fig. 5, take any point *u*. From *u* draw the radii vectors *y*, *y'*. Now, *bA* is the length of pitch line, or length of semi-major axis. We know that the sum of the focal radii, or radii vectors, is equal to twice the length *a*, or to twice the length of pitch line. As $\frac{P}{2\pi}$ locates the

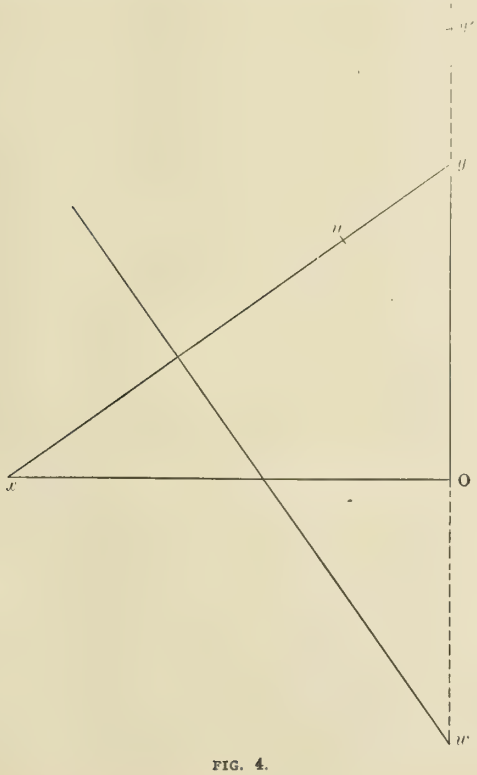


FIG. 4.

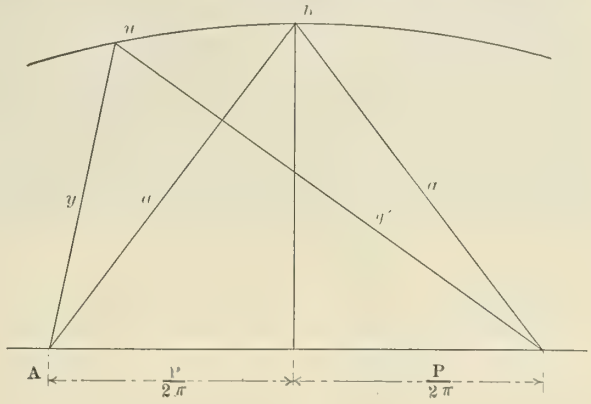


FIG. 5.

foci of all elliptic arcs about *Y Y'*, Fig. 1, and *y*, *y'* are focal radii, or radii vectors, as above; therefore, $y + y' = 2\ a$.

The writer has been frequently asked why, when a blade has a fall back, should not the developed area be raised above the projected area? In other words, why should not the developed area be drawn as though *Q Z* were swung around *Z* until it was at right angles to the axis of the shaft. This would make the contour of blade show above its projected view.

If two straight generating lines, such as *OZ* and *OZ'*, Fig. 6 (*OZ* being at right angles to the axis, and *O Z'* inclined to *O Z* at any angle), move together, generating screw surfaces, the form and pitch of both being equal, the helices of the two surfaces so formed must be exactly similar. Now, these surfaces will always be at a constant distance from each other, and for any radius such as *Ba*, Fig. 1, represented by *r* in Fig. 6, this distance will, therefore, be $Ba \times \tan \Theta = r \times \tan \Theta$, where Θ is the angle of inclination of the generating line.

Therefore, when *O Z'* at any radius leaves the surface, *O Z* at the same radius leaves its surface. Therefore the surface

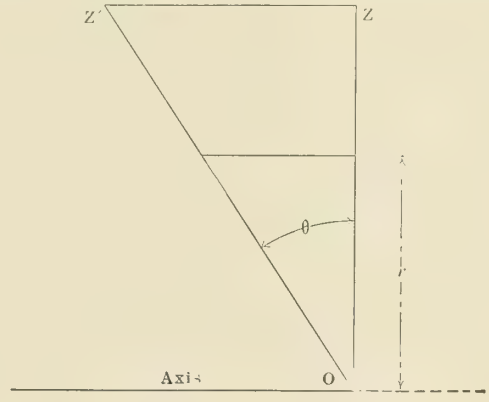


FIG. 6.

formed by *O Z* will be precisely the same as that formed by *O Z'*. Taking Fig. 1, for instance, the developed area formed by *Q Z* will be precisely the same as that formed by *Y''' Z*. We see, therefore, that a blade generated as above is obtained simply by setting the helices at intersected distances aft, corresponding to points selected. This can be further proved by laying down a template, and fitting the same to the pattern. This shows that the athwartship projection for the same developed area is independent of the fall-back or skew.

(To be Concluded.)

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Attack on the Florida.

On May 27, the American single-turret monitor *Florida* was attacked by her sister ship *Arkansas,* one 12-inch shell, filled with high explosive, being directed against the face of the turret, close to the starboard gun, while one 12-inch and four 4-inch shells were fired at an experimental mast, built up completely of spirally-placed steel tubes to a height of about 90 feet.

The object of the attack was to determine the effect of the detonation of a high explosive shell against a modern turret, as regards both penetration and the disturbance of electrical and other fittings within the turret. The second part of the attack was intended to determine the value for fire-control purposes of such a mast as was fitted to the *Florida* for this purpose. In neither case was the attack or its object at all similar to those carried out by the British Admiralty against the old ironclads *Belleisle* and *Hero,* in which cases the object appears to have been to riddle the ships and effect a complete destruction. The attack differed in another way, in that the two British ships were condemned and of no further use, while the *Florida* is still serviceable as a coast-defense vessel, having in fact been launched as recently as 1900.

While the detailed results have not been made public, enough is known for us to state that after the attack on the turret, the turret turning, gun elevating and ammunition hoist mechanism was still in perfect order, and readily responded when called upon to function. The attack on the mast resulted in severing several of the tubes of which it was built, but such was the strength of the entire structure and its stability against failure, even in such a wounded condition, that it was still perfectly serviceable, and could have withstood apparently a much more severe fire than that to which it was subjected.

It is proposed to make a further experiment by mooring the *Florida* in shallow water and directing against her side a modern torpedo. The object of this is to determine to what extent inner subdivision may resist the explosion of such a torpedo, special compartments having been built into the ship for this purpose.

Marine Engine Design.

In this issue is being started a serial on the above topic which, it is expected, will run throughout the balance of the calendar year. The intention is to republish this in book form as soon as the serial is concluded.

Most of the text-books on the subject of marine engines deal only in a general way with the subject of design, and particularly with this subject as referring to the numerous co-ordinating parts of the modern marine engine. No matter how extensive the work, nearly all of the space is taken up with descriptions and illustrations of engines of various types, with some regard to historical features, but with very little to the actual work of laying out and designing the engine. It is felt that the little work at present under way will fill a decided void in this respect, and, being totally free from the descriptive part of the subject, which may be obtained in any text-book now on the market, it will be much more readily available for the particular use for which it is intended.

It is not the idea in this work to cover in detail the design of every part of the engine; but enough is given in the way of detailed design of the principal parts to indicate the general scope of the problem, and to lay down methods by which the entire work can be carried to completion. As given, this represents the result of several years of experience in teaching the subject of marine engine design to students of the University of Michigan. Irrelative material has been carefully excluded, and the whole subject is given in a concise and thoroughly readable manner.

Boilers and auxiliary machinery have been omitted

completely; the former forming a separate subject, and, indeed, under the present increasing steam pressures, we find that special watertube boilers are rapidly taking the place of their firetube predecessors, and these watertube boilers, being constructed by various manufacturers under their own patents, would scarcely form a fruitful subject for the discussion here of design. Auxiliary machinery is also largely in the hands of builders of special types, and comes under the same general classification with regard to the desirability of attempting to design it in a work of this sort. The propelling machinery alone is considered, but this is given in such detail as to make a fairly complete, though very brief, treatise.

Propeller Wheels.

Two articles this month deal with the ever-vexing subject of the propeller. One of them, from the pen of a practical marine engineer, discusses the laying out of propeller wheels, and is to be followed next month by a treatment of the making of patterns for propellers. The other article is the reprint of a paper read by an expert investigator before the Institution of Naval Architects, and is also to be concluded next month.

We have frequently had occasion to refer to this subject editorially, usually in connection with some article or series of articles which we were publishing. It is one of the utmost importance, and one to which, in most quarters, too little attention is paid. More than this, it is a subject which is perhaps as far from being an exact science as anything in modern shipbuilding. No two designers could be expected to turn out identical propellers from a given set of conditions and requirements; and frequently of half a dozen dissimilar designs, actual trials would have to be made to determine which was best suited for the purpose in hand. This arises from the fact that different features connected with the design of a propeller are differently evaluated by the various designers, and that the subject, so far as the laying down of specific rules for design is concerned, is still in a very unfinished state. It is for this reason that every bit of new information which can be brought to bear is of value, whether that information be the result of the trials of actual propellers on ships for which they were designed, or the result of experiments conducted with model propellers, either with or without an attendent hull.

For obvious reasons it is usually impracticable to run a model propeller in connection with a hull. In the first place, the models are usually run in a series, showing variation along one or more of their component elements, such as pitch, blade area, etc., and it would be practically out of the question to furnish hulls in sufficient number and variety to fit each of these separate cases. In addition to this, to make a complete test with the propeller accompanied by the hull, it would be necessary to arrange for both single and twin screws, and, indeed, if the practice is to apply to our latest turbine-driven ships, three and four screws would also have to be provided for. Not only this, but variation would necessarily be required with regard to the clearance between the tip of the revolving propeller blade and the hull, and, in the case of the single-screw ship particularly, the location in a fore-and-aft direction would introduce still another element of variation.

For this reason, nearly all of our available material in the shape of accumulated data from propeller experiments has been acquired through the running of propellers in free and undisturbed water, without any reference to the presence or absence of any hull or other structure. There has, of course, necessarily been such other structure present, but it is always located behind the propeller, and in such a position as not to affect the operation of the latter. This structure is required for the proper housing of that part of the mechanism designed to rotate the propeller.

Because of this divorcing of the propeller and the hull which it has to propel, these model experiments have been condemned in some quarters as totally unreliable. It cannot for a moment be claimed that the results of such an experiment would be the same without the hull as with it. Experience has shown, however, that for determining and delineating the action of the propeller, pure and simple, model experiments are not only trustworthy, but of very great value.

Perhaps the most notable instance of the application to practical use of the experimental model propeller occurred in connection with the building of the first vessel ever propelled by a steam turbine. This was the *Turbinia,* built in 1894. Originally this vessel was fitted with one large propeller. Instead of the high speed hopefully anticipated, only about half the ultimate speed was developed with this propeller. This means that the useful work done by the propeller was perhaps not more than 10 percent of that ultimately obtained by the fitting of the propellers with which the vessel made her first phenomenal burst of speed. Small propellers on three shafts were afterwards fitted, but there was an unaccountable loss of power between the boiler and the propulsion of the ship. Investigation by means of model propellers showed very clearly that this loss was due very largely to a phenomenon not unknown at that time, but much better known since, and which is called cavitation. It has been observed in many instances in connection with the operation at very high speeds of revolution of propellers designed for the absorption of considerable power. The studies which Mr. Parsons undertook at this time, added to those made by Mr. Barnaby, led to a complete change in the policy of fitting propellers for high-speed vessels; with the result that speeds of 30 knots and upwards are now attained regularly, and with little difficulty, by vessels of varying types and sizes, and with propellers so designed as to minimize the ill effects of this phenomenon.

Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, reports the following percentages of completion of vessels for the United States navy:

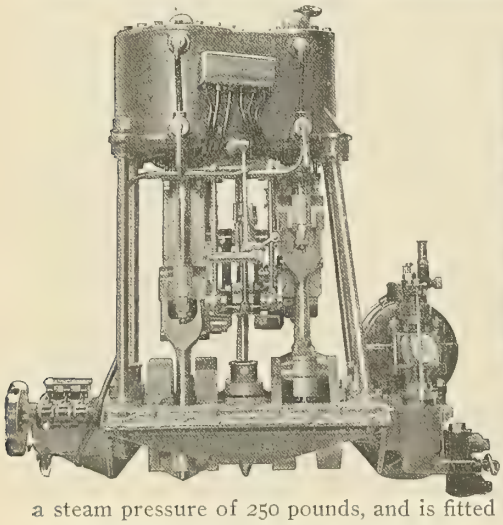
	Tons.	Knots.		April 1.	May 1.
BATTLESHIPS.					
South Carolina..	16,000	18½	Wm. Cramp & Sons.....	42.2	45.9
Michigan.....	16,000	18½	New York Shipbuilding Co....	48.6	50.7
Delaware.....	20,000	21	Newport News S. B. & D. D. Co	18.1	22.8
North Dakota...	20,000	21	Fore River Shipbuilding Co....	25.7	31.6
ARMORED CRUISER.					
Montana.....	14,500	22	Newport News Co.....	97.	98.
SCOUT CRUISER.					
Salem.....	3,750	24	Fore River Shipbuilding Co....	95.2	96.4
TORPEDO BOAT DESTROYERS.					
Number 17.....	700	28	Wm. Cramp & Sons.....	12.	15.9
Number 18.....	700	28	Wm. Cramp & Sons.....	10.8	15.4
Number 19.....	700	28	New York Shipbuilding Co.....	11.5	16.2
Number 20.....	700	28	Bath Iron Works.....	8.	9.7
Number 21.....	700	28	Bath Iron Works.....	7.8	9.3
SUBMARINE TORPEDO BOATS.					
Cuttlefish.....	—	—	Fore River Shipbuilding Co....	99.	99.
Number 13.....	—	—	Fore River Shipbuilding Co....	30.	40.5
Number 14.....	—	—	Fore River Shipbuilding Co....	30.	39.9
Number 15.....	—	—	Fore River Shipbuilding Co....	30.1	34.
Number 16.....	—	—	Fore River Shipbuilding Co....	29.9	34.2
Number 17.....	—	—	Fore River Shipbuilding Co....	10.3	13.6
Number 18.....	—	—	Fore River Shipbuilding Co....	10.3	13.4
Number 19.....	—	—	Fore River Shipbuilding Co....	10.3	13.

ENGINEERING SPECIALTIES.

The Savery Steam-Yacht Engine.

The engine illustrated was fitted by T. A. Savery & Company, Newcomen Works, Birmingham, into a fast passenger launch for use on the Nile. The launch has a length of 40 feet, a beam of 8 feet, a draft of 2 feet 9 inches, and a speed, loaded, of about 12 miles per hour.

The engine is a quick-running compound of 38 indicated horsepower, with cylinders 4 and 8 inches diameter by 5 inches stroke. It operates at 620 revolutions per minute under



a steam pressure of 250 pounds, and is fitted with an outboard or keel condenser. Steam is supplied from a quick-steaming watertube boiler fired with petroleum fuel, in which steam can be raised in a few minutes from cold water.

The weight of the machinery outfit complete, with stern gear, pipes, etc., is only 2,100 pounds, while the over-all length is just under 6 feet 3 inches. On account of the high rate of revolution, the air and feed pumps, which are worked off the main engine, are geared down to a reduced speed to insure efficient and silent operation.

Particular attention has been paid to the balance of moving parts, especially in the valve gear and connecting rods. There is a noticeable absence of vibration and smooth turning effort. The Joy valve gear is used, while all glands and bearings are of ample size and length.

A Small Speed Counter.

There has always been a demand for a small vest pocket speed counter which is accurate and durable. The speed counter here illustrated is one that is said to contain these qualities. It is of the same size as the illustration, being not

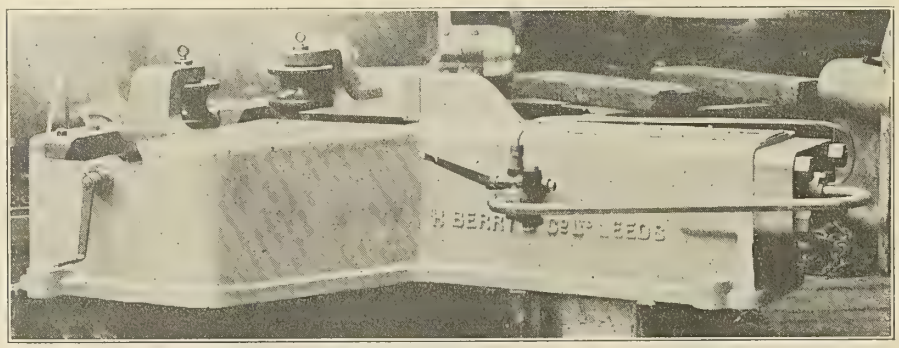
quite 3 inches in length. While it is very small and convenient to carry it is by no means fragile. It registers the exact number of revolutions, requiring only the taking of the initial reading, and subtracting it from the final reading, to obtain the number of revolutions in time elapsed.

This unique speed counter has been placed on the market by the American Steam Gauge & Valve Manufacturing Company, 206 Camden street, Boston, Mass.

A Pipe-Bending Machine.

The photograph illustrates one of the latest types of pipe bending machines made by Henry Berry & Company, Ltd., Croydon Works, Hunslet, Leeds. This machine is designed principally for the purpose of bending steel and copper pipes, and is invaluable in marine engineering shops, where a large amount of this class of work has to be done.

The machine is fitted with two adjustable sliding blocks carrying rollers or bollards. These blocks are adjustable by means of screws, worked independently from either side, so that unequal sided bends can be made. The head is fixed on the hydraulic cylinder, which, having a long travel, can pass right through the adjustable sliding blocks for bending U or expansion joints.



The principal advantage claimed for this machine is that the obstructions on the top of the frame are reduced to a minimum, and so interfere as little as possible with the work being done. By the use of suitable blocks, angles, bars, channels and joists can be either bent or straightened.

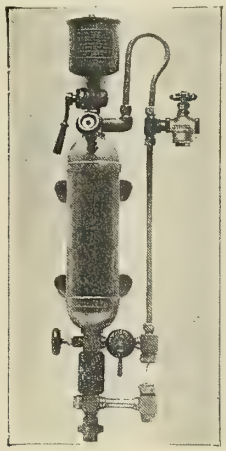
The machine illustrated has a power of 45 tons, and is capable of bending pipes up to 6 inches diameter. It was supplied to the order of the Admiralty for the workshop on Portland breakwater.

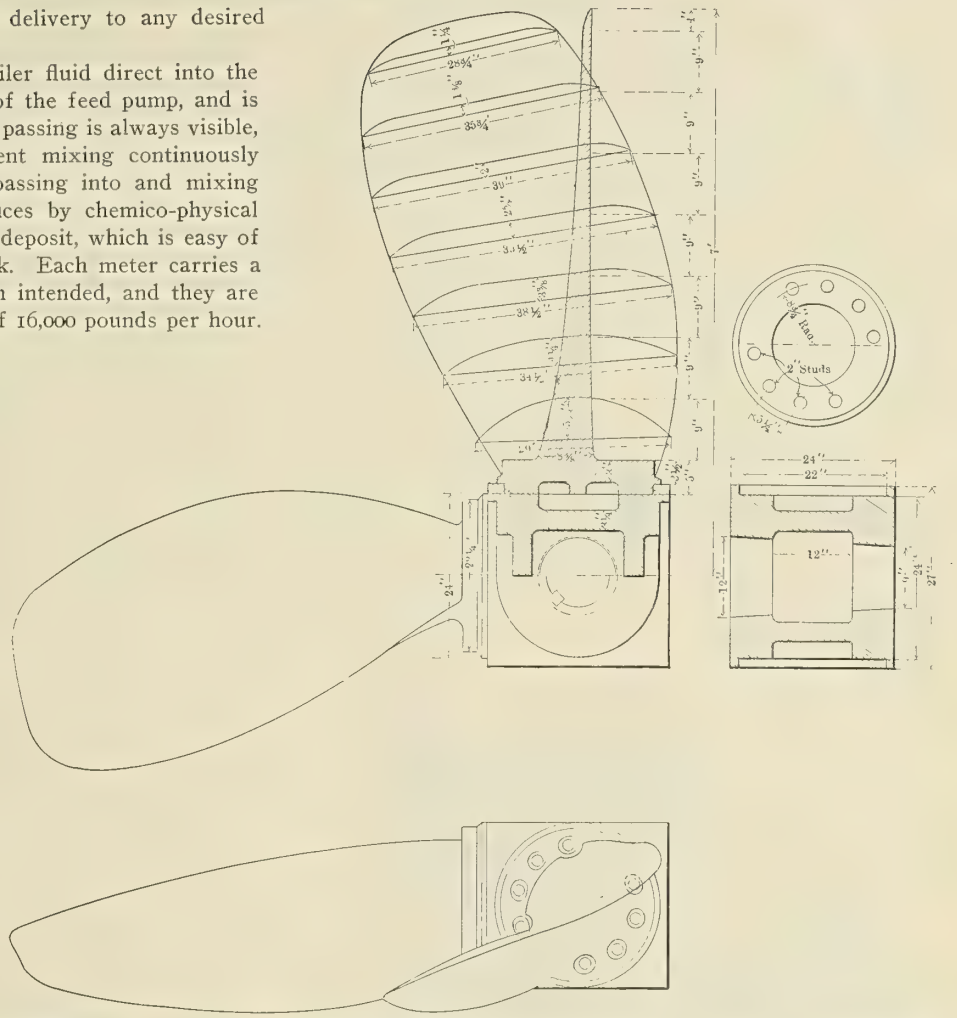
An Automatic Sight-Feeding Meter.

The "Boilerine" sight-feeding meter is placed on the market by the Boilerine Manufacturing Company, London, S. E., for the purpose of limiting incrustation in a boiler, thereby prolonging its life, reducing the expense of up-keep and saving fuel. A gunmetal casting with hemispherical ends, tested to 500 pounds per square inch, is fitted with a valve and an indicator fixed to the valve body, and a pointer by means of

which the operator can adjust the delivery to any desired speed.

The meter feeds the Boilerine boiler fluid direct into the boilers drop by drop, independently of the feed pump, and is so arranged that the quantity of fluid passing is always visible, this method guaranteeing the reagent mixing continuously with the feed water. The liquid, passing into and mixing with the water in the boiler, produces by chemico-physical action a loose non-adhesive granular deposit, which is easy of removal by means of a blow-off cock. Each meter carries a day's supply for the boiler for which intended, and they are made in sizes up to an evaporation of 16,000 pounds per hour.





A Successful Type of Propeller.

The propeller illustrated was built by H. G. Trout, King Iron Works, Buffalo, N. Y., and is four-bladed, with detachable blades and adjustable pitch. The diameter is 14 feet; the pitch, as designed, 17 feet; the developed area, 64 square feet, and the disk area, 154 square feet. The ratio of disk to developed area is 2.4 to 1.

This is one of a series of propellers built by the Trout Works, the wheels being made from stock patterns, over 400 in number, for all sizes from 1 to 15 feet in diameter for solid wheels, and more than seventy patterns from 3 to 15 feet in sectional wheels.

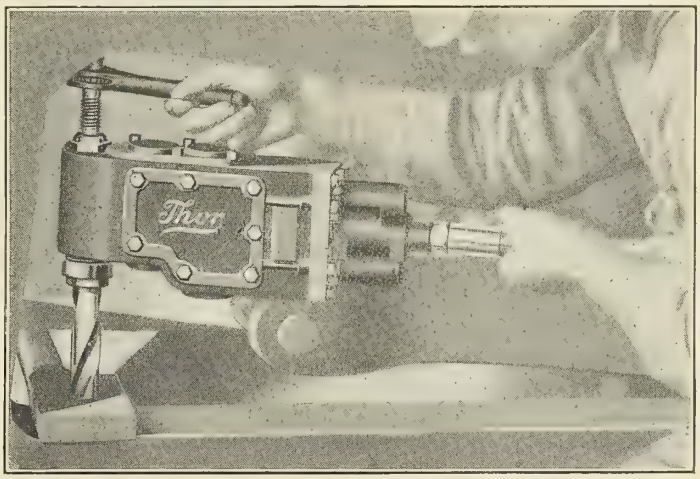
Thor Close-Quarter Pneumatic Drill.

The Independent Pneumatic Tool Company, of Chicago and New York, has just placed on the market a close-quarter piston air drill, the dimensions of which are as follows:

	Inches.
Throttle connection to outside of spindle case	15⅜
Point of feed screw to end of socket	8⅞
Radius from center of feed screw to outside of case	1 9/32
Width of case at cylinder flanges	5 3/16
Width at spindle	6¼

Weight, 31 pounds; speed, 122 revolutions per minute.

The spindle is at one extreme end of the tool, and the motor is at the opposite end. The motor consists of two cylinders parallel with each other, and at right angle to the spindle, the center line of both cylinders centering on the center of spindle. The pistons are double acting, and operate on a two-throw crank. Between the crank throws at the center are located the eccentrics, the cranks and eccentrics being one forging. The eccentric straps operate directly on balanced cylindrical piston valves, having a reciprocating motion. The air is taken in centrally between the cylinders, and the valves control the air as close to the cylinder bore as



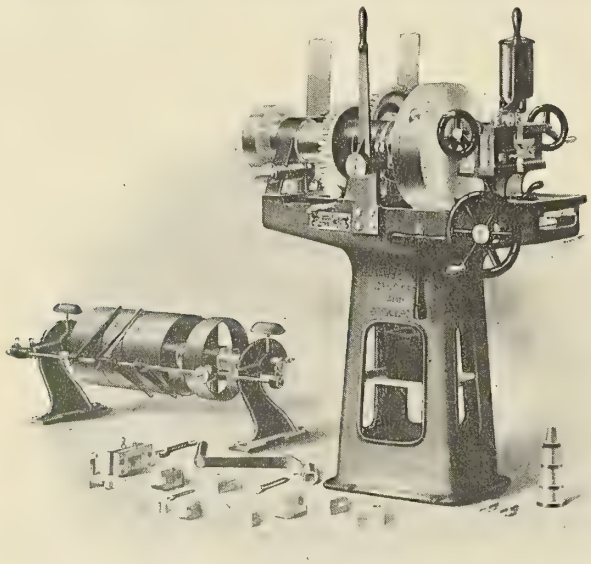
material will permit. The engine crank proper is not on the usual 90-degree angle, but has an angle of 135 degrees, thus allowing two pistons to pull when the position of levers requires the greatest power. This makes the drill in a degree self-regulating, and tends to still further govern the speed of the entire revolution of drill spindle. This drill is provided with reversible ratchet feed mechanism, operated within the width of the body of the drill itself. A poppet valve throttle controls the speed and power to a nicety, and also acts as a handle.

A New Pipe Machine.

This machine is placed on the market by the Crane Company, Chicago, to meet the demand for a low-priced machine, operated by hand or power, for high-class service. All parts have been designed to withstand any strains that such a pipe cutting and threading machine may be subjected to. Simplicity of operation, adjustment and arrangement has been carried out to the minutest detail.

This tool possesses many features which increase output and facilitate ease of operation. The gripping, threading, cutting-off and adjustment have been so arranged that no unnecessary operations are required. The frame is one casting, having bed and stand in one piece, eliminating the use of light legs, and giving greatest rigidity with minimum weight and floor space.

The die head is bolted to a movable carriage, with ample travel. Upon the die head are the dies, pipe guides and cutting-off tool. The dies are of the improved adjustable type, made collapsible. They are carried in suitable frames, sliding in guides, and moved by a screw operated by a hand wheel. They are set to gage by a simple locking device, which allows any number of pieces of pipe of the same size to be threaded without any further adjustment. These dies have four cutting



edges, and will give good service on either steel or wrought iron pipe. Dies are made interchangeable, and one die of a set may be replaced if broken, thus reducing the repair bill to the minimum.

When cutting off, the pipe is guided by two steel guides, hardened on the face. These guides are operated by a right and left screw and hand wheel. The cutting-off tool is operated by a lever and rack. This makes a rapid, simple and positive device and extremely powerful. The gripping chuck is of the quick gripping type, rapid in action and very powerful. Pipe may be released and gripped by the throwing of a lever without stopping the machine. The chuck is adjustable to the different sizes of pipe within range of the machine, without moving or altering the jaws. The jaws are of tool steel carried in steel holders, and are removable for grinding or replacing.

The capacity is from ⅛-inch to 2-inch pipe. The dimensions of the countershaft pulley, running at 200 revolutions per minute, are 9¾ inches diameter by 3½ inches face. The floor space required is 44 by 23 inches; the weight, 700 pounds.

The *Mauretania* recently made 635 nautical miles in one day's steaming westward. The *Lusitania* promptly answered with 641.

TECHNICAL PUBLICATIONS.

Warships: A Text-Book on the Construction, Protection, Stability, Turning, Etc., of Warships. By Edward L. Attwood, M. I. N. A. Size, 6 by 8¾ inches. Pages, 316. Figures, 230. London and New York, 1908: Longmans, Green & Company. Price, 10/6 net and $3.00 net.

This is the third edition of a work first issued in 1904, intended primarily for the use of naval officers, and as a text book covering the designing of the principal features of construction of warships. It is divided into twenty-four chapters and two appendices, followed by an index, and takes up in rotation the strength of ships, tests of steel, sections, rivets and joints, the framing of various types of ships, beams, pillars and decks, plating of the outer and inner bottoms, watertight bulkheads and doors, stems, sternposts, rudders, shaft brackets and steering gears, pumping, flooding, drainage and ventilation, corrosion and fouling, coaling, armoring and deck protection, mensuration rules, navy list displacement and tonnage, buoyancy, tons per inch, stability, trim and stability at large angles of inclination, the rolling and turning of ships, the resistance and propulsion of ships, the design of warships and notes on the loss of H. M. S. *Victoria*.

The work is eminently practical, and gives details of the structural arrangements of most of the principal types of battleships and large cruisers in the British navy up to the *King Edward VII.* and *Lord Nelson* classes. The arrangement of armor is shown, with the forms of backing of wood, and the framework upon which the armor rests. The chapter on the design of warships is quite general, outlining the method of procedure in designing such vessels, and giving in some cases distributions of weights. Notable in this connection are the weights given of five Japanese war vessels, the battleship *Asahi*, the armored cruiser *Asama*, the cruisers *Takasago* and *Akashi*, and the destroyer *Akutsuki*, as well as that of a small cruiser of 2,650 tons, belonging to the British navy. The first appendix consists of a large number of questions under each chapter, with answers thereto. The second appendix is a memorandum of the main elements of the *Dreadnought* and *Invincible* designs, presented to Parliament in 1906.

While it would be impossible in the scope allowed by 300 pages to even outline the designing of an entire warship, the principal features are taken care of in this book in such a manner as to cover the main points of construction of the hull and its general fittings. No attempt is made to go into engine or propeller design, nor are the military features, aside from the armoring of the hull and barbettes, given any attention whatsoever. The distribution of guns and their protection are left for the designer to figure out for himself, the work being, in fact, concerned simply with the hull design of warships in general and of battleships and large cruisers in particular.

Night Signals of the World's Shipping. By D. H. Bernard. Size, 5½ by 8½ inches. Pages, 127. Numerous illustrations in colors. Glasgow, 1908: James Brown & Son. Price, 7/6 ($1.75 net).

This book is a compilation of the night signals of the various steamship lines and yacht clubs, and is intended to be such an aid to the officer on watch as to enable him to recognize at once a vessel's signal, thus avoiding much delay and confusion. By previous methods it has been necessary to turn over many pages of printed details before finding a signal, whereas in the present case the colors are arranged in order, and in such a manner as to be readily accessible.

The work is divided into eleven sections, the first five of which are devoted to pyrotechnic and other lights under the several headings of blue, green, red, white and yellow, each color mentioned being the predominant one of the signal. The other sections include rocket displays; roman candle displays, separate and accompanied by other lights; sound signals, with or without lights; guns, and Lloyd's signal. Blank

spaces are left throughout the book for additional signals to be noted, although the number given is very great and covers practically all that would be necessary in most cases.

Structural Engineering. By Prof. A. W. Brightmore, D. Sc., M. I. C. E. Size, 5½ by 8⅜ inches. Pages, 280. Figures, 145. London, E. C. (La Belle Sauvage), and New York, 1908: Cassell & Company. Price, 10/6 net and $3.75 net.

This work is designed to fill the need for a text-book suitable for students of engineering, and placed intermediate between the subject of strength of materials and such specialized works as bridge construction, etc. With this idea in view, the endeavor has been to treat in a consecutive and intelligible manner the principal ideas and methods underlying the investigations necessary in the designing of structures. The historical side has been omitted, but certain aspects of the subject, such as the methods of construction and the use of the equilibrium polygon and the stress ellipse, have been given rather more prominence than is usual in works of this sort.

It is recognized that in most engineering problems it is not possible to make hypothetical assumptions and convert them into mathematical equations. As a result, mathematics have generally to be applied in detail to each part of a problem, and the assumptions readjusted at each step, to prevent the mathematical deductions from diverging to any considerable extent from actual conditions. The success of the design of any engineering structure depends largely upon the judgment of the engineer, based on a knowledge of the fundamental principles of statics.

The work is divided into fifteen chapters, followed by an index, and is devoted mainly to the design of girders, bridges and arches, together with retaining walls, foundations, dams and structures of reinforced concrete. The illustrations are all of the line type, and are sufficiently clear for the purpose intended. The typography is very good.

Machine Design, Construction and Drawing. By Henry J. Spooner, C. E.; professor of mechanical and civil engineering in the London Polytechnic School of Engineering. Size, 5⅜ by 8½ inches. Pages, 691. Figures, 1,424. London and New York, 1908: Longmans, Green & Company. Price, 10/6 net and $3.00 net.

This is a text-book for the use of students and young engineers, and in it the work is carried through in regular rotation, beginning with a description of drawing instruments and their uses and various methods for testing them.

The first five chapters are devoted to these elementary considerations, after which are taken up stuffing boxes, collars, shafting, cranks, journals, couplings and clutches, keys and pins, riveted joints, bolts, nuts and screws, pipes and connections, cotters, knuckle joints, bearings and hangers, roller and ball bearings, toothed and friction gearing, belts, rope and wire rope gearing, chains, crane hooks, tanks, pistons and rods, crossheads, guides and crank rods, eccentrics, springs, and two large chapters devoted to miscellaneous items, such as materials used in construction, strength of beams, drop forgings, data regarding pumps, etc., etc.

The work is very profusely illustrated with sketches and occasional half-tones, and is well fitted out with tables giving proportionate parts and constants for working. Examples are given for examinations, and the whole work is so arranged as to be readily applicable for use as a text-book. Many exercises, especially at the ends of chapters, are provided with answers where such can be expressed in figures. Much of the information used has been drawn from the technical press of England and America, as well as some works from the continent of Europe.

Equipment Buyers' Finance: The Effect of Speeds on Profits. By Arthur Winder. Size, 8½ by 11¼ inches. Pages, 16. Leeds, 1908: Electric Printing Works. Price, 1s. net.

This is a brief discussion in five chapters of the subject of machine tools as affected by the type used and the speed of operation. The subject of using an old lathe of low first cost, as compared with a high-speed lathe of higher cost, is taken up, and the latter is shown to be more economical when it comes to comparing output with cost.

QUERIES AND ANSWERS.

Questions concerning marine engineering will be answered by the Editor in this column. Each communication must bear the name and address of the writer.

Q. 407.—In a triple-expansion engine with cylinders 20, 36 and 57 inches in diameter by 24 inches stroke, the revolutions per minute are 150; the steam pressure, 165 pounds gage; the cut-off in high-pressure cylinder occurs at three-fourths stroke; the clearance is 10 percent. A single feed pump of the plunger pattern is connected to the engine, and has a stroke of 10 inches. What should be the plunger diameter?
2.—How many times the theoretical quantity of water would you decide on for the capacity of the pump? P. G.

A.—1. The formula commonly used in designing feed-pump plungers is:

$$D = \sqrt{\frac{550 \times Q}{N \times L}}$$

where D is the diameter of the plunger in inches; L is the stroke in feet; N is the number of working strokes per minute, and Q is the quantity of feed water in cubic feet per minute. On this basis

$$D = \sqrt{\frac{550 \times 2.13}{150 \times 0.83}} = \sqrt{9.372} = 3.06 \text{ inches.}$$

This calls for an area of plunger of 7.35 square inches.

The above computation was made on two assumptions. In the first place, the indicated horsepower of the engine was assumed, from the data given, to be 500; in the second place, the consumption of steam per horsepower-hour was assumed to be 16 pounds. This gives 8,000 pounds of steam per hour, or 133 pounds per minute, which amounts to 2.13 cubic feet per minute. The number of working strokes of the pump is, of course, the number of revolutions of the engine per minute, because the pump, being of the plunger type, is single acting.

2. The displacement of a plunger of the area and stroke given is 0.0425 cubic foot per stroke, or 6.375 cubic feet per minute. This is 398 pounds per minute of water, theoretically. It will be noticed that this is almost exactly three times the required amount of water for the engine, which is explained in two ways. In the first place, it is customary to fit a feed pump of twice the required capacity, in order to be able when necessary to fill the boiler rapidly. In the second place, leakages in the pump and the necessity for making up a certain amount of loss of feed, due to leakages in the engines and connections and to the occasional use of steam for whistling, require that the plunger displacement be considerably greater than the theoretical figure. These two considerations have operated to make the plunger area about three times the size theoretically called for.

Q. 409.—How may the number of gallons of water discharged from an orifice in a given time be computed, when the pressure of the water and the size of orifice are known?
2.—Is there a formula giving the ratio of water pressure to the velocity of the water? D. F. B. L.

A.—The formula is

$$Q = C \sqrt{2gH} \times A,$$

in which Q is the number of cubic feet per second; g is the gravitation factor, or 32.16 feet (per second)^2; H is the head of water measured in a vertical line between the center of the orifice and the level of still water above it; A is the area of the orifice in square feet. The factor C, which is almost universally taken as 0.62, takes account of the fact that water, in flowing through an orifice, has a tendency to take on a

contracted cross-sectional area, due in large measure to the friction against the sides of the orifice. To reduce the quantity to gallons the number of cubic feet would be multiplied by 7.84 (for imperial gallons, multiply instead by 6.24). The velocity of the water is represented by the expression $\sqrt{2gH}$. The static pressure at center of orifice is H.

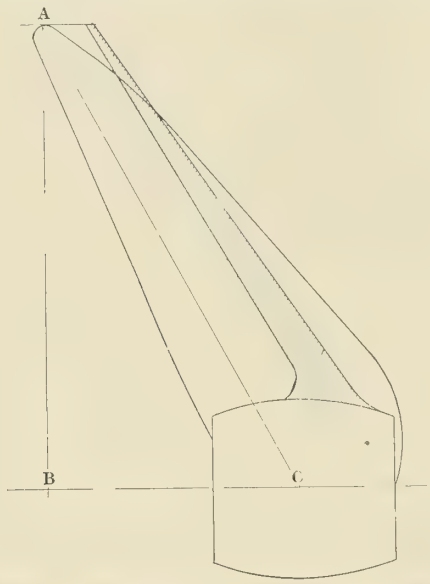
To take a concrete example, suppose we have a square orifice measuring 2 feet on the side, and with its center 4 feet below the surface of still water. Then

$$Q = 0.62 \sqrt{2g} \times 4 \times 4 = 40 \text{ cubic feet per second.}$$

This represents about 300 United States gallons, or 250 British imperial gallons, per second. The velocity is 16 feet per second. The mean static head is 4 feet or a pressure of 1¾ pounds per square inch.

Q. 408.—How can I find the area of a propeller blade while the ship is in drydock?
2.—Having a blade with an excessive droop astern, will the diameter be equal to twice the radius on line *A B*, or on line *A C*? JUNIOR.

A.—1. We assume that the developed area is what is wanted. By drawing a straight line down through as nearly as possible the center of the blade from hub to tip, and setting off perpendiculars from this line at known distances apart, the lengths of these perpendiculars can be spaced off with dividers and set down on a drawing. The developed area of the blade may thus be traced in through the outer ends of these lines, and the area obtained either by means of a planimeter or by some process of approximate integration. For the latter it would be necessary to have the perpendiculars equally spaced from root to tip. In this case *the summation of all the lengths, added to one-twelfth the sum of the second and the next to last, and diminished by seven-twelfths of the two end ordinates*, would give a factor which, multiplied by the distance between ordinates or perpendiculars, would give the



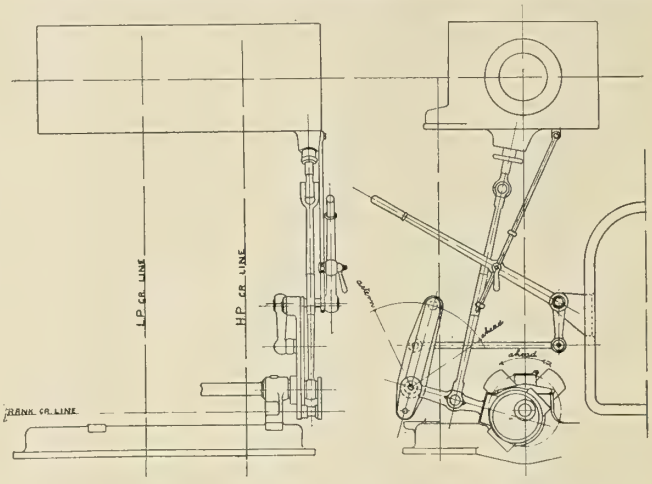
desired result. This would not include the approximately triangular sections between the perpendicular at the root of the blade and the root itself as it follows the contour of the hub. For practical purposes, however, the area in this section of the propeller is of very little value, particularly as regards thrust, and for computations. If it were desired, however, an approximation sufficiently close for the purpose could easily be made.

The natural way to take ordinates of this sort would be, not along a perpendicular to the center line, but along arcs of concentric circles, of which the common center is the center of the propeller shaft. For the case in question, however, this might not be feasible, and the method above outlined will give the result equally well.

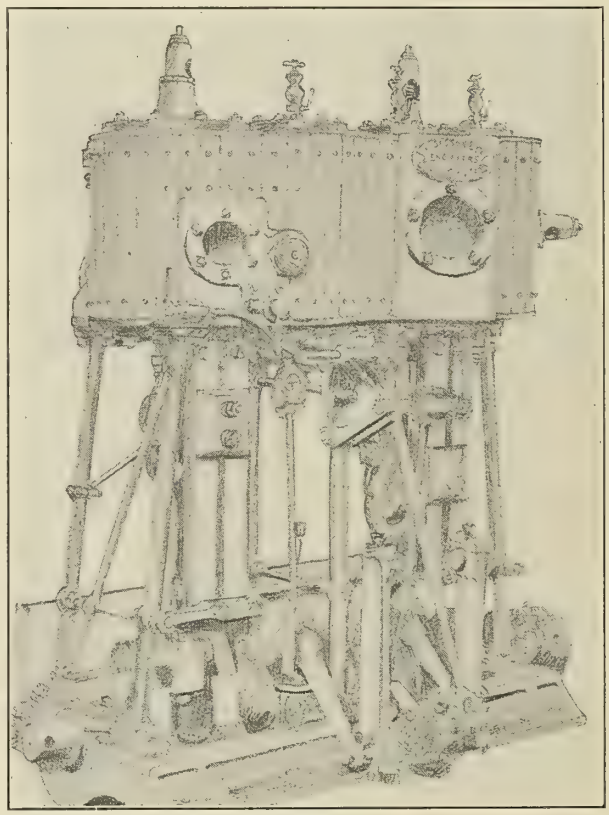
2. The diameter would be twice the radius *A B*. The diameter of a propeller wheel is defined as the diameter of the circle swept by the tips of the blades while in rotation.

Q. 406.—On page 94, in your February number, in describing a small steam-yacht engine, built by W. Sisson & Company, Ltd., Gloucester, you state that the valve gear is of their specially designed, single-fixed eccentric or elliptic type. Please give an outline sketch and the name of this gear. K. I.

A.—The sketch shows the arrangement of the gear as fitted



to a compound surface-condensing engine, with cylinders 9 and 15 inches in diameter and 10 inches stroke. This gear is Messrs. Sisson's modification of what has been commonly known as the Marshall valve gear, which itself was a modifi-



cation of the original Hackworth gear. The latter had a slide, the angle of which could be varied in relation to the crank shaft, while the Marshall or elliptic valve gear has a swing link. The inclination of the path of the eccentric lever end is varied by moving the weigh shaft in or out, thus giving the necessary obliquity and the elliptic movement to the lower end of the intermediate valve rod.

Personal.—By an act approved April 16, concerning the revenue cutter service of the United States, the ranking officer of the engineer corps (Chief Engineer Charles A. McAllister) is continued as engineer-in-chief, with a rank raised from that corresponding to captain in the army to that of lieutenant-colonel. Under the same act the commandant of the service (Captain Worth G. Ross) is raised in rank to that corresponding to colonel in the army or captain in the navy.

William T. Donnelly, consulting engineer, has moved from 132 Nassau street to 135 Broadway, New York.

Frederick D. Herbert, formerly editor of this journal, is in charge of the New York office of the Terry Steam Turbine Company, at 90 West street.

Obituary.

Rear Admiral Charles W. Rae, engineer-in-chief of the United States navy, and chief of the Bureau of Steam Engineering of the Navy Department, died in Washington, May 13. He was born in Hartford, Conn., in 1847, and was graduated from the Naval Academy in 1868. He was appointed engineer-in-chief and chief of the bureau on Aug. 3, 1903, for a term of four years, which was renewed last August for a similar period. He has been succeeded by Captain John K. Barton, who becomes engineer-in-chief, and head of the bureau.

Samuel Samuels, skipper of the *Henrietta,* which won the first transatlantic yacht race, and for nine years captain of the famous clipper *Dreadnought,* died May 18. With the *Dreadnought,* in 1859, he made a record across the Atlantic from England to New York of 9 days 17 hours, actually beating by some 24 hours the steamship records of those days. He was president of the *Marine Journal* Company, in which office he has now been succeeded by the editor, Capt. George L. Norton.

Francis B. Stevens died in Hoboken May 23, at the age of 93. He was a civil and mechanical engineer of great note during the first half of the nineteenth century, his best known invention being the Stevens cut-off, first applied to the steamboat *Albany* in 1840, and in use on practically all side-wheel steamers of the present day. He was prominently identified with the introduction of railroads into the United States.

A. S. Crowninshield, rear admiral, U. S. N., died in Philadelphia, May 27, at the age of 65. He was a graduate of the Naval Academy at Annapolis, and saw service in the civil war. During the Spanish war, and for some years thereafter, he was chief of the Bureau of Navigation in the Navy Department, and was a member of the Naval War Board.

SELECTED MARINE PATENTS.

The publication in this column of a patent specification does not necessarily imply editorial commendation.

American patents compiled by Delbert H. Decker, Esq., registered patent attorney, Loan & Trust Building, Washington, D. C.

881,803.—PROPULSION OF SUBMARINE BOATS. GEORGE F. JAUBERT, PARIS, FRANCE.

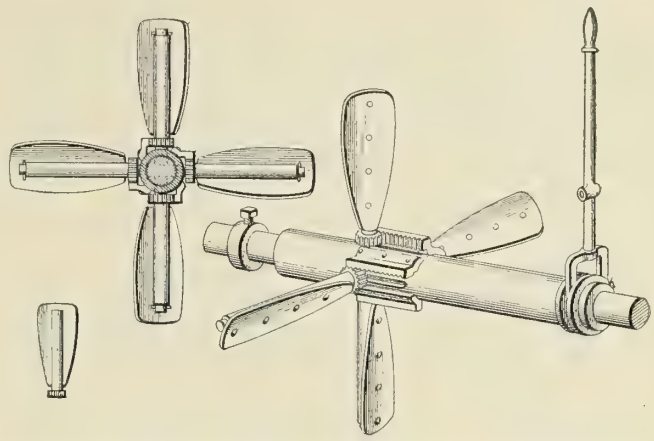
Abstract.—According to this invention, an explosion gas engine is fed in a closed cycle; *i. e.,* without air being utilized for the combustion of the fuel. The combustion gases exhaust into a purifier or washer, where the steam is condensed and the carbonic acid partly absorbed, so that an inert gas is obtained, which is then mixed with oxygen and is capable of being again utilized in the engine for producing an explosive mixture with the liquid fuel. Five claims.

884,936. APPLIANCE FOR ADDING TO THE RECORD SPEED OF VESSELS. WILLIAM LAUDAHN, LOS ANGELES, CAL.

Abstract.—The invention relates to a system of pipe or other conduits which are placed upon the exterior of the submerged portion of the hull, said pipes leading from a tank or reservoir placed within the hold of the vessel. The tank or reservoir contains an oil, emulsion or like liquid having little affinity for water. The oil, etc., by means of the apparatus, is ejected in minute quantities and distributed to the submerged portion of the hull, thereby materially reducing the friction, in addition to preventing the hull from fouling. Two claims.

881,775. SCREW PROPELLER. ROSCOE E. COON, PORTLAND, OREGON.

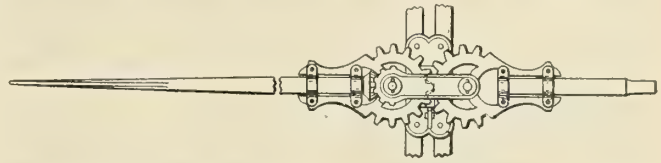
Claim 2.—The combination of a propeller shaft having rigid right



angular arms, propeller blades with sockets and gear wheels on the inner ends of the same mounted to turn on said arms, an outer sleeve inclosing the propeller shaft and having longitudinal slots to give passage to said arms, rack bars arranged beside the slots and fixed to said sleeve, means for sliding said sleeve longitudinally over the propeller shaft to adjust or reverse the pitch of the propeller blades, and stops for limiting the throw of the blades. Three claims.

882,072.—OAR. JOSHUA LEHMAN, TORONTO, CANADA.

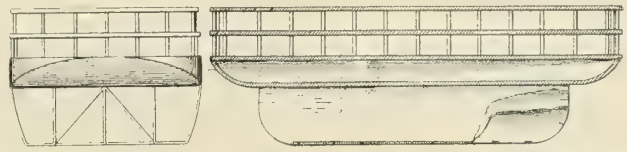
Claim 1.—In an oar, in combination, a pivotal block supported from the gunwale of a boat and having pins extending upwardly from each end thereof, a pair of co-acting segmental toothed gears journaled on said pins, an oar handle rotatably supported from one of said gears, an oar stem rotatably supported from the other of said gears, a bevel pinion fixedly secured to the inner end of said oar stem and rotating



therewith, and a plate having orifices therethrough, through which pass the pins extending from the pivotal block, said plate being rigidly secured to the pins above the segmental gears, and having a bevel gear surface formed on its under side at one end, and meshing with the bevel pinion secured to said oar stem, and adapted to rotate said pinion on the swinging of the segmental gear supporting said oar stem, to feather the oar.—Three claims.

882,194.—BOAT. JOHN W. GAY, RICKREALL, OREGON.

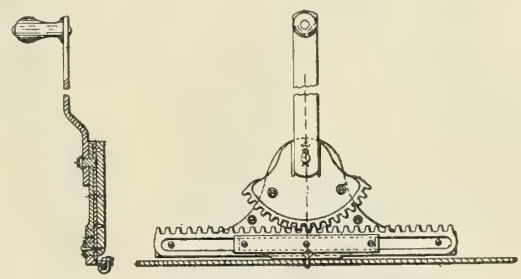
Abstract.—The boat consists essentially of an air chamber upon which its floating capacity depends. The top of this air chamber, being always above the waterline, constitutes the main deck of the vessel.



Above this main deck are other decks, having either open or inclosed sides, as may be necessary. Below the air chamber, the boat is formed with longitudinal compartments of special construction arranged for the free longitudinal passage of water through them, but at the same time acting as ballast means to prevent a tipping of the boat. Four claims.

882,821.—STEERING DEVICE FOR POWER BOATS. JAMES A. GARRETT, AUBURN, N. Y.

Claim 4.—In a power boat having a steering wheel, a tiller and a tiller rope, a supplemental steering mechanism connected to the tiller



rope at a point between the steering wheel and the tiller, having in combination a supporting plate, a toothed segment rockable thereon, means to rock the segment, a rack-bar meshing with the segment, a slot in the rack-bar, and a slide support in said slot. Six claims.

883,423. SHIP-CLEANING APPARATUS. ARTHUR R. ROGERS, JONESPORT, MAINE.

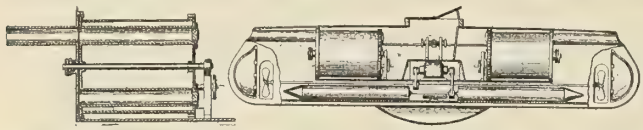
Claim 5.—The combination with a guiding frame constructed to engage the keel of a vessel, and ropes attached to said frame, of a series of revolving scrapers free at one end and connected at their other ends with the guiding frame. Nine claims.

883,588.—INVISIBLE AIR-CHAMBER AND SPONSON FOR CANOES. CALEB B. THATCHER, BANGOR, MAINE.

Claim.—A canoe having tapering curved ribs cut to conform to the curvature of the hull secured to the outside of the planking, and extending from the gunwales to the waterline, said ribs gradually decreasing in width and length from the center towards each end, planking secured on said ribs, and canvas secured over said hull and ribs, having its side edges secured under the gunwales of the canoe, forming an invisible airtight compartment. One claim.

883,664. TORPEDO BOAT, ERNST A. NILSEN, CHRISTIANIA, NORWAY, ASSIGNOR OF ONE-HALF TO BERNARD CRAFTON, CHICAGO.

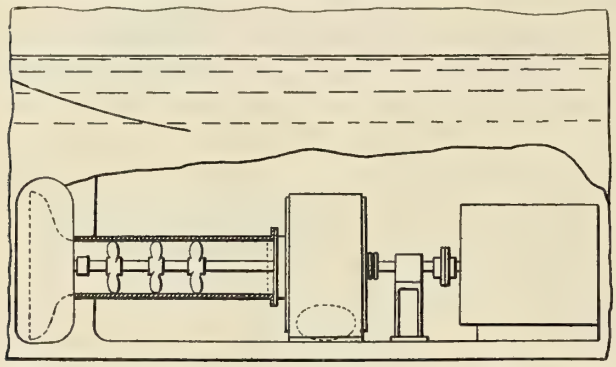
Claim 2.—In a torpedo boat, or submarine, the combination with a torpedo launching tube opening outwards through the hull of the boat, of a rotary magazine drum provided with a plurality of torpedo



chambers adapted to register with said launching tube, hand operated means for rotating said drum, and hand operated means for releasing said torpedos from said chambers when desired, comprising a hand rod with a yoke pivoted thereto and adapted to bind against said torpedo when in its forward position, and to engage the starting arm of the torpedo when drawn rearward. Four claims.

884,079. PROPULSION OF SHIPS. EDWARD J. DUFF, LIVERPOOL.

Claim.—Means for propelling ships, comprising a prime mover, a centrifugal pump actuated thereby, a shaft therefor, a turbine mounted

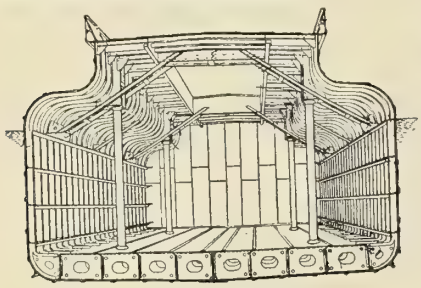


directly on said shaft and in the lower part of the boat, means connecting the turbine with the water outside the boat whereby the power which is in the head of water required to float the ship is utilized, said centrifugal pump adapted to eliminate the back pressure on the turbine, and capable of removing more water than is admitted by the turbine. One claim.

British patents compiled by Edwards & Co., chartered patent agents and engineers, Chancery Lane Station Chambers, London, W. C.

25,165.—SHIPS' FRAMES AND FRAMING. C. D. DOXFORD, SUNDERLAND.

Turret vessels are fitted with vertical columns or supports arranged upon the line where the inswept frames turn upwards to form the turret. The columns are preferably of tubular construction, secured at their lower ends to the hollow bottom, and at their upper ends to the frames, by brackets or their equivalents. Light diagonal struts are also fitted, and extend from the frames to the beams. The struts are arranged in pairs, one on each side of each column. Deep frames strengthened by horizontal stringers are used in the ship's construction, if found necessary. In a modification, in which "'tween decks" are fitted, the columns are attached to such decks, and smaller columns are fitted between the decks and the beams.



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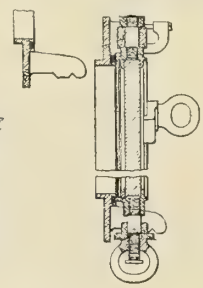


25,455

25,455.—SHIPS' CABIN LIGHTS. J. BROADFOOT & SONS, WHITEINCH, and J. R. APPLEBY, PARTICK.

Relates to that type of ship's side scuttles in which the scuttle ring is pivoted in a block having side trunnions working in bearings in lugs on the scuttle frame. A stud is fitted in the scuttle ring, and has a square block mounted on its unscrewed part, which extends between lugs on the frame. In the edges of these lugs are two notches in which projections engage. The scuttle ring is held in its closed position by any convenient fastenings, and may be held slightly open by permitting the projection to engage the catch, the ring turning on trunnions. The trunnions are mounted in lugs having slots, so that the ring may be turned at right angles to the frame.

24,854. PROPELLING SHIPS AND BOATS. W. COCHRANE, LONDON, W.

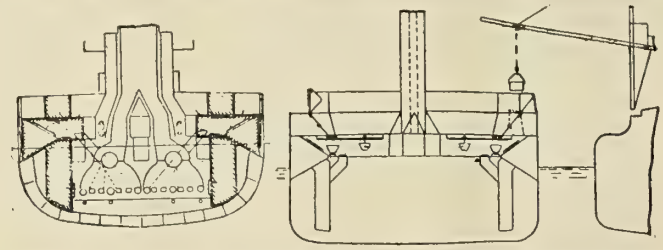
The propellers are of the vibrating-plate type and are arranged in pairs, operated by motors, the boat being fitted with two bilge keels. Each motor has a shaft, extending on opposite sides of the crank chamber, and provided with a clutch. A disk fixed obliquely to the shaft is grooved and carries on ball bearings a strap, to which is pivoted a fork on the vertical shaft of the propeller. The amplitude of oscillation of the propeller may be varied by altering the obliquity of the disk; to attain this object, the disk is pivoted to the end of the shaft, and is linked by a rod to a sliding collar, which is controlled by a pivoted lever or by means of worm gearing. To prevent wear of, and noise due to, the propeller, the vanes are beaded to engage a socket in the propeller frame, the end of the frame having an overhanging flange, so that the water is prevented by the flange from escaping, and acts as a buffer.

24,879. SHIPS' CABIN; BERTHS. A. H. BAIRD, LIVERPOOL.

Relates to the metal fittings by which the portable partitions of staterooms and berths for ships are secured to the upright posts forming the framework of such rooms and berths. To avoid the gap below the joint caused by the necessity for having a clearance space, the fitting is provided with a sliding piece. A hook engages a cup attached to the post, and the sliding piece fits closely against the cup and is there maintained in position. As the hook and the partition fall into place, relative movement takes place between the cross-plate of the fitting and the sliding piece. The flanges on the latter cover the gap which would otherwise be formed below the cup.

25,506.—WARSHIPS' BUNKERS. DUISBURGER MASCHINENBAU AKTIEN GESELLSCHAFT, FORMERLY BECHEM & KEETMAN, DUISBURG, GERMANY.

Coal is stored partly in hold bunkers and partly in lower-deck

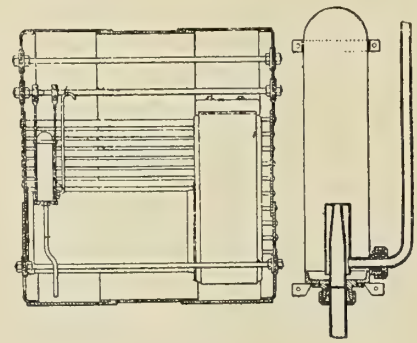


bunkers, and instead of the lower-deck bunkers acting as reserve bunkers to the hold bunkers, they are provided with shoots for discharging the coal directly into the stokehold, without passing it through the hold bunker. In one example, the lower-deck bunkers are divided longitudinally by a partition provided with doors, and are fitted with shafts for charging, which are closed by other doors. The hold bunkers are filled through shafts and doors, and are discharged into the stokehold by other doors. The lower-deck bunkers communicate directly with the stokehold by means of shoots. In a modification, the part of the deck and the side plating outboard of the shaft are hinged to one of the decks, and are moved to facilitate charging the various bunkers. Suspension truckways may be fitted in the lower-deck bunkers.

26,249.—SHIPS' STAIRCASES; VENTILATING. F. ALCOCK, BIRKENHEAD.

The stairs or ladders used to gain access to the various holds of a ship are disposed within a tube which extends through the decks. Openings and landings are provided at each deck, and sliding doors are fitted at convenient places. The tube may also be utilized as a ventilating pipe, and for that purpose is provided with a cowl at its upper end. The tube may be constructed of openwork, such as bars or gridwork, fixed between the decks.

26,249



26,967

26,967. STEAM GENERATORS. J. BRUNDRIT, LIVERPOOL.

Water-circulation, promoting.—The boiler is fitted internally with a closed vessel and connected by pipes with the steam space and the lower part of the water space. The air trapped in the vessel during the filling of the boiler expands when heated by the surrounding water, and drives out the water through a pipe. Water then enters the vessel through another pipe. The first pipe is connected with a casing surrounding the conical end of the latter pipe and open at the top and bottom. The vessel may be fitted in the smoke-box and may have cross-tubes; or it may be composed of tubes.